SUPPRESSION OF HYPERBARIC CHAMBER NOISE

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by

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ABSTRACT

The air noise levels presently found in the U.S. Navy's hyperbaric chambers are unacceptably high when compared to the standards established by the Walsh-Healy Act.

This paper demonstrates a means of reducing this air noise found in hyperbaric chambers. A muffler was designed, built and tested in a model of a hyperbaric pressure chamber. The muffler design was analyzed using an electrical-acoustical analog and the network analyzing program MARTHA. A second muffler design based on the first design results was built and tested. The results of the analysis and testing of both designs are discussed and recommendations for future noise suppression efforts are made.

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DEFINITIONS OF SYMBOLS

ACFM	Cubic feet per minute flow rate measured at ambient conditions in the chamber
C	Electrical network element for capacitance
C_A	Acoustical compliance
FSW	Feet of seawater (0.445 psi=1 ft seawater) Describes depth of diver or equivalent pressure of chamber
IL	Insertion Loss, dB
L	Electrical network element for inductance
M_A	Acoustical mass
P_{ga}	MARTHA network parameter-power available from generator
P_o	Barometric pressure in newtons/m^2
P_{out}	MARTHA network parameter-power to the load
R	Electrical network element for resistance
R_A	Acoustical resistance
SCFM	Cubic feet per minute flow rate measured at standard conditions (70°F, 14.7 psia)
SPL_{band}	Sound-pressure level measured with bandwidth Δf, dB
S(f)	Spectrum level, dB
TG	MARTHA network parameter-Transducer Gain
TL	Transmission Loss, dB
U	Volume velocity in m^3/second
V	Enclosed volume of air with opening for air pressure variation
Z	Electrical network element for impedance

ZG	MARTHA network parameter-generator impedance
ZL	MARTHA network parameter-termination impedance
a_1	Radius of inlet pipe in meters
a_2	Effective radius of holes in the baffle plates-design one
a_3	Effective radius of outlet plenum-both designs
a_4	Radius of 4" pipe
a_5	Effective radius of holes in baffle plates A-A and C-C of design two
a_6	Effective radius of holes in the 4" pipe of design two
a_7	Effective radius of holes in baffle plate B-B of design two
a_8	Effective radius of outlet chamber of design two
c	Speed of sound in air in m/second
dB	Decibels (A reference pressure of 0.00002 newton/m^2 is used)
kl	$\frac{2\pi l}{\lambda}$
l'	Effective length of component including end effects
l''	Effective length of component including end effects
m	$\frac{\text{Cross-sectional area of the chamber}}{\text{Cross-sectional area of the duct}}$
p	Pressure in newtons/m^2
w	Radians per second

γ	Ratio of specific heats for air= $\frac{c_p}{c_v}$
λ	Wavelength of sound at the temperature of the gas
μ	Kinematic coefficient of viscosity of air at standard temperature and pressure
ρ_o	Density of air at standard temperature and pressure

CHAPTER I

INTRODUCTION

Hyperbaric chambers are intended for the treatment of the adverse physical effects that may occur during underwater diving operations. When a diver loses his air supply underwater, he has an overwhelming instinct to hold his breath and come to the surface immediately. The lack of adequate exhalation during a panicky ascent creates excessive pressure in the lungs because the diver has been breathing compressed air underwater. If the diver should come to the surface holding his breath, this air will expand and rupture his lungs, allowing air to enter his blood vessels and cause obstruction of blood flow to his brain. This condition is termed an air embolism. Decompression sickness, also called caisson disease, or the bends, results from inadequate decompression following exposure to any inert gas at a critical depth and for a critical time. Bubbles of the inert gas are formed in the tissue and blood stream and by mechanical obstruction cause pain, paralysis, asphyxia and, if large or numerous enough, can be fatal. Both air embolism and the bends are treated by placing the diver in a hyperbaric chamber (Figure 1) and recompressing the expanded gases in his body by pressurizing the chamber. The diver is treated at various depths for times as specified by treatment tables.

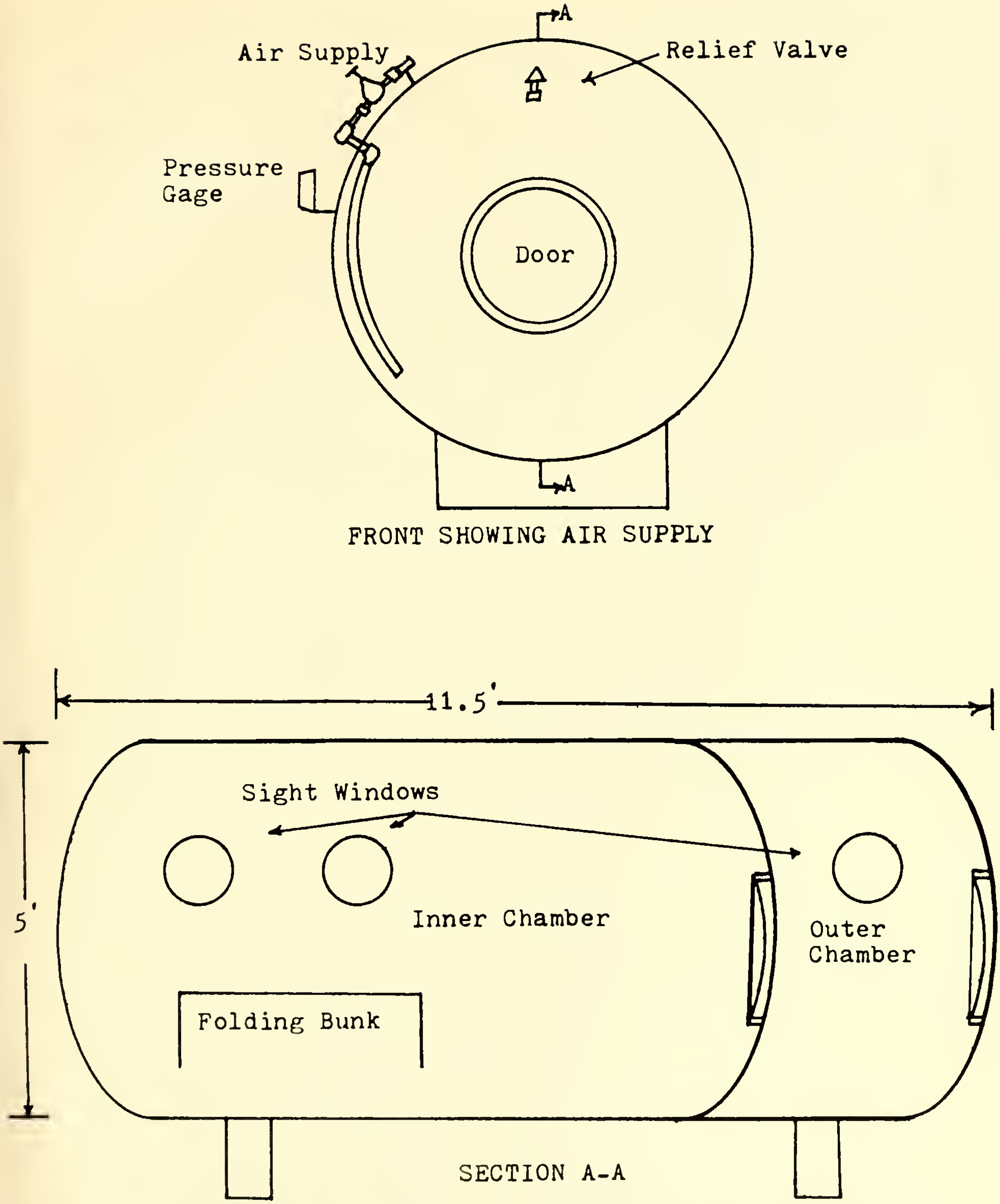


Figure 1-Simplified Hyperbaric Chamber showing inner and outer chambers and air supply line.

Hyperbaric chambers are commonly used as training devices for neophyte divers to determine their reaction to the effects of increased pressure. The chambers are also useful for pressure testing new diving equipment.

The noise levels encountered during the pressurization, ventilation and depressurization of hyperbaric chambers are high enough that they are considered a possible contributor to the hearing deficiencies of divers(1,2). * The noise levels also complicate the evaluation of heart and breath sounds of a diver undergoing treatment in a hyperbaric chamber(2).

This thesis is an attempt to reduce the noise levels in hyperbaric chambers. Theoretical design methods and experimental results are used to design, test and evaluate an air noise muffler for the air supply system of hyperbaric chambers.

* Numbers in parenthesis refer to the reference list.

CHAPTER II

AIR NOISE IN HYPERBARIC CHAMBERS

During compression of a hyperbaric chamber the in-rushing air raises the noise level to a point that conversation is useless and the ears are left ringing. When the chamber has reached the desired depth the air is stopped and the chamber is relatively quiet. Within a short period (2 to 5 minutes) it becomes necessary to ventilate the chamber and the noise starts again. This time the noise results from a combination of new air entering the chamber and old air exhausting. The pressure in the chamber is maintained at as constant a level as is possible during ventilation. When returning to atmospheric pressure the noise level again rises as the air is exhausted.

The operational procedures for using hyperbaric chambers for treatment of numerous symptoms are established in the USN Diving Manual(7). Duration and depth of stops for the various treatments are defined in the manual. The rates of descent and ascent and the ventilation rates required are also clearly established in the manual. As an example, the descent rate (increasing pressure) is usually 25 feet of seawater per minute (0.445 psi = 1 ft seawater). The continuous ventilation rates are 2 ACFM (ft^3/minute at ambient conditions in the chamber) for each patient breathing

air, 4 ACFM for a tender breathing air, 12.5 ACFM for each patient breathing pure oxygen and 25 ACFM for a tender breathing pure oxygen. In Table 1 these quantities are equated to standard ventilation rates (SCFM) for each of the various treatment depths in which two patients and one tender occupy the chamber (8). It is presently not possible to ventilate continuously at these rates since the noise level is too high during ventilation for conducting medical examinations or carrying on conversations. The present practice is to ventilate intermittently at higher rates of air flow. Intermittent ventilation is not nearly as effective in reducing levels of unwanted gases as is continuous ventilation. The amount of pressurized air used when continuously ventilating is less than that used for intermittent ventilation (8).

Recognizing the problems brought about by hyperbaric chamber noise, a number of investigators have measured the noise levels (1,2,4,5,6). Comparing the measured levels with the standards as outlined by the Walsh-Healy Regulations of May 1969 emphasizes the need for suppression of hyperbaric chamber noise.

The Walsh-Healy regulation specified 90 dB on the A-weighted scale of the sound level meter for 8 hours a day as the level above which hearing damage may occur. The A-weighted scale, which closely represents the sensitivity of the human ear to the effect of noise, is well suited

DEPTH OF STOP (FSW)	TREATMENT TABLES 1-4 VENT RATE (SCFM)		TREATMENT TABLES 5-6A VENT RATE (SCFM)	
	AIR STOP	O_2 STOP	AIR STOP	O_2 STOP
165	47.9	----	47.9	----
140	41.9	----		
120	37.0	----		
100	32.2	----		
80	27.3	----		
60	22.5	140.7	22.5	140.7
50	20.1	125.6		
40	17.7	110.5		
30	15.3	95.4	15.3	95.4
20	12.8	80.2		
10	10.4	65.0		

Table 1-Ventilation Flow Rate Requirements for Treatment Tables 1-6A of the USN Diving Manual (Two Patients and One Tender in Chamber) (Reference 8)

for measuring noise levels relevant to hearing loss(2). The United States Department of Labor in May 1969 accepted and published in the Federal Register those noise levels at which every industry in the country under the Walsh-Healy regulation must administer a continuous and effective hearing conservation program. The Bureau of Medicine and Surgery under the Department of the Navy outlined the current hearing conservation program for the U.S. Navy(3) and established the total duration of noise exposure allowable (Appendix A) at levels based on the Walsh-Healy regulation.

Summitt and Reimers(1) at the Navy Experimental Diving Unit measured hyperbaric chamber sound levels at various depths during compression at approximately 60 feet per minute and while ventilating the chamber at a constant depth. The results are shown in Table 2 as equivalent A-weighted sound levels corrected for increased ambient pressures.

Description of the Environment	Gas Supply Valve Setting	Depth in Feet of Seawater 0	50	100	150	200
Hyperbaric Chamber Ventilating	Full Open	116 dBA	121 dBA	118 dBA	118 dBA	116 dBA
Hyperbaric Chamber Descending	Sufficient for Compression Rate of 60 FPM	-	116 dBA	114 dBA	110 dBA	107 dBA

Table 2 A-weighted sound levels obtained in the hyperbaric chamber during routine operations(1).

Further measurements made at the Experimental Diving Unit(6) of sound levels in a ventilated chamber indicate that noise levels increase slightly as depth is increased but the frequency spectrum is unchanged (Figure 2).

At the Naval School, Diving and Salvage (NSDS), Harvey(2) conducted audiometric examinations on ten experienced divers before, during and after exposures to 41.4 psi (60 ft) chamber pressure while breathing air. The noise levels during descent, ascent and venting were monitored. Harvey states that the maximum equivalent A-weighted sound level of 108 dB (corrected for pressure) during venting offers a significant risk of hearing damage to subjects exposed to hyperbaric chambers for long periods. Uncorrected maximum noise levels recorded during descent were 106 dBA and during ascent were 108 dBA.

At the Naval Submarine Medical Center Murry(4) measured the noise levels during the descending and ascending stages of two dives to 100 feet. The maximum A-scale reading obtained was 120 dB during the descending stage and 115 dB during the ascending stage.

In an effort to reduce the measured noise levels, tests were conducted at the Experimental Diving Unit(5) using two filter elements as mufflers. The filter elements were screwed onto the air supply line discharge. Sound levels were measured for various settings of the air supply valve, both with and without the filters. The

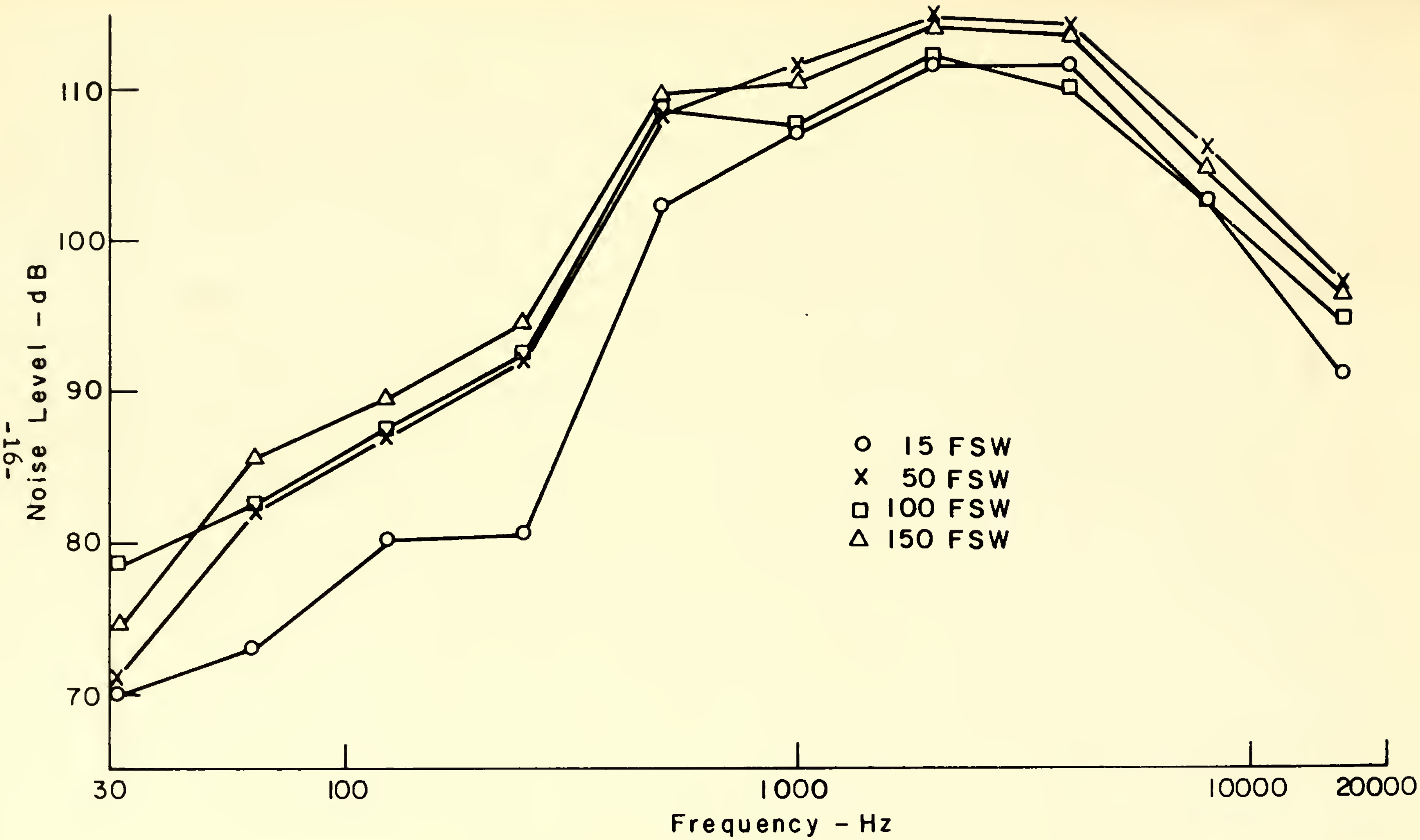


Figure 2—Noise Spectra of Ventilating Experimental Diving Unit Chamber.

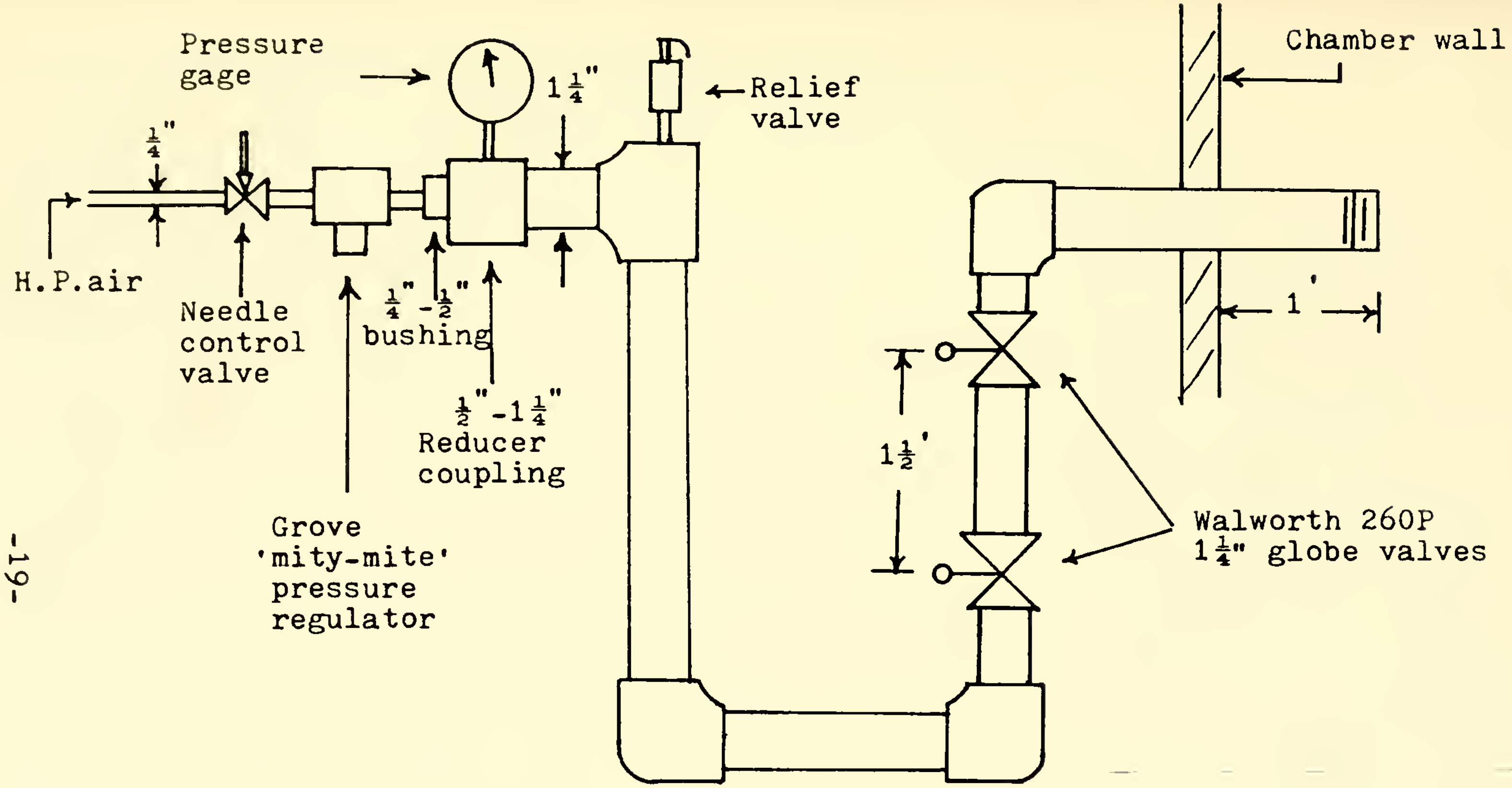
tests were all run at one atmosphere total pressure, with a 250 psig supply pressure to the valve. A 30 dBA drop in the measured sound levels was created by the filters. The filter elements themselves are combustible and are therefore not suitable for use in hyperbaric chambers. The results of this test indicate that the high noise levels existing in present chambers can be easily reduced using an all-metal acoustic silencer or muffler(5).

CHAPTER III

MODELING THE CHAMBER AIR SYSTEM

The first step in muffler design is to determine the noise characteristics of the system without a muffler It is best to have actual measurements of the system noise, or noise spectra of the components that generate noise. Once the actual noise spectrum has been found, the next step is to decide upon the maximum noise spectrum that is acceptable with the filter installed. The differences between the unmuffled and the acceptable noise spectra establishes the minimum insertion loss that the muffler must provide as a function of the frequency. The final step in the design of the muffler is to make experimental noise measurements with the muffler in place. As a result of these noise measurements modifications may be necessary to achieve the desired insertion loss (16).

The air supply system of one of the hyperbaric chambers at the Navy School Diving and Salvage(NSDS) was modeled at the Gas Turbine Laboratory at the Massachusetts Institute of Technology. The Gas Turbine Lab has a high pressure air system storage capacity of 34.3 cubic feet. The maximum system pressure is 2800 pounds per square inch. An air supply was constructed as shown in Figure 3. Figure 4 is the air supply system for a hyperbaric chamber at NSDS.



<u>Figure 3</u>-Air Supply System Constructed as Model of Hyperbaric Pressure Chamber. The total length of low pressure piping is 20.6 feet.

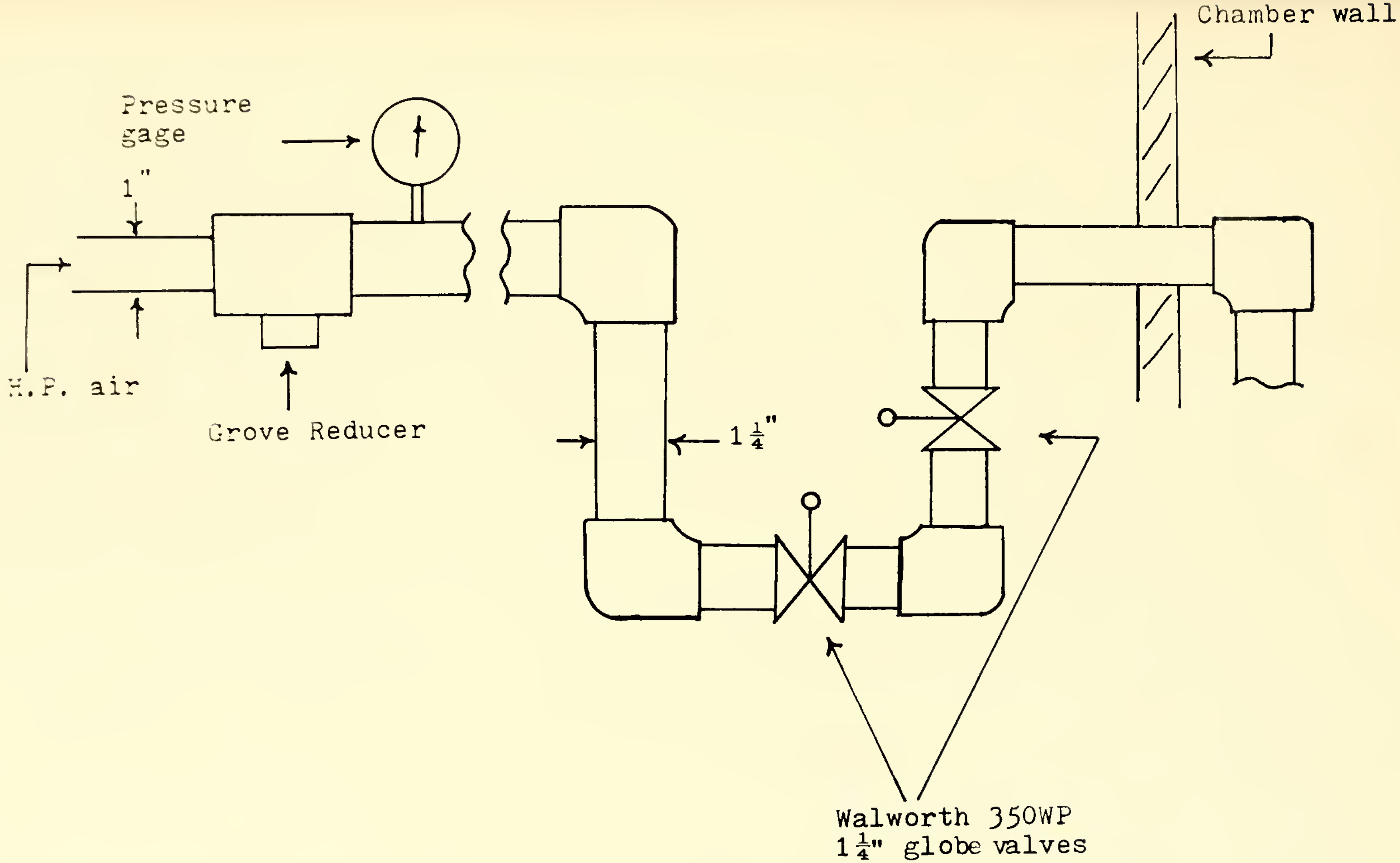


Figure 4-Simplified Air Supply System for a Hyperbaric Chamber at NSDS. The total length of low pressure piping is 46 feet. The distance between the two 1 1/4" globe valves shown is 4'10". (Reference 14)

The NSDS system has a much greater capacity than the model but the noise characteristics are similar at the flow rates of interest (140 SCFM and lower). In both systems the low pressure piping is $1\frac{1}{4}$ inch ID and the two inlet valves are Walworth globe valves. It is important that the model valves be similar to the chamber valves since their noise-making characteristics influence the noise spectra.

The chamber at the Gas Turbine Lab is a cylinder 12 feet long and 6 feet in diameter. The air supply was brought into the chamber 1 foot from the forward door and 3 feet from the bottom. The air exited the chamber through a 7 inch diameter hole in the forward door. At no time was the chamber at any than atmospheric pressure. The microphone was located 2.5 feet from the opening of the air supply line at a height of 2 feet from the floor and 6 inches from the forward door. This is about the proper position for a diver's ear when seated and leaning against the door. The microphone was not situated directly in the airstream of the air entering or leaving the chamber.

The noise was measured using a 1/8 inch Brüel and Kjaer condenser microphone in conjunction with a Brüel and Kjaer frequency analyzer type 2107. This instrument is a constant percentage bandwidth audio frequency analyzer. Using this frequency analyzer it is possible to obtain one third octave band readings of the noise spectra (11). Using one third octave band readings allowed the spectrum to be

analyzed in a relatively short period of time and yet provided the necessary accuracy for muffler design and testing. The frequency analyzer is also able to measure the sound level using an A-weighting network.

Measurements of the noise level in the chamber were taken with the pressure regulator set at 300 psig. The upstream $1\frac{1}{4}$ inch globe valve was used as the throttling valve with the other one fully open.

The noise generated by the air supply system of the model was measured in runs one and two. In run one both valves were fully open and an average flow rate of 145 SCFM resulted. The A-weighted sound level measured was 118 dB. The third octave band noise spectra of the model is shown in Figure 5 for runs one and two. The noise spectrum of the model is compared with that of the actual hyperbaric chamber in Figure 6. The points for the model spectrum were computed from run two by use of the following equation:

$$SPL_{band} = S(f) + 10 \log_{10} \Delta f \quad \text{dB}$$

where SPL_{band} = sound pressure level measured with bandwidth Δf

Δf = bandwidth, Hz

$S(f)$ = spectrum level, dB

The points for the actual chamber spectrum were those shown in Figure 2 as measured at a pressure of 15 feet.

The details of these initial runs as well as those

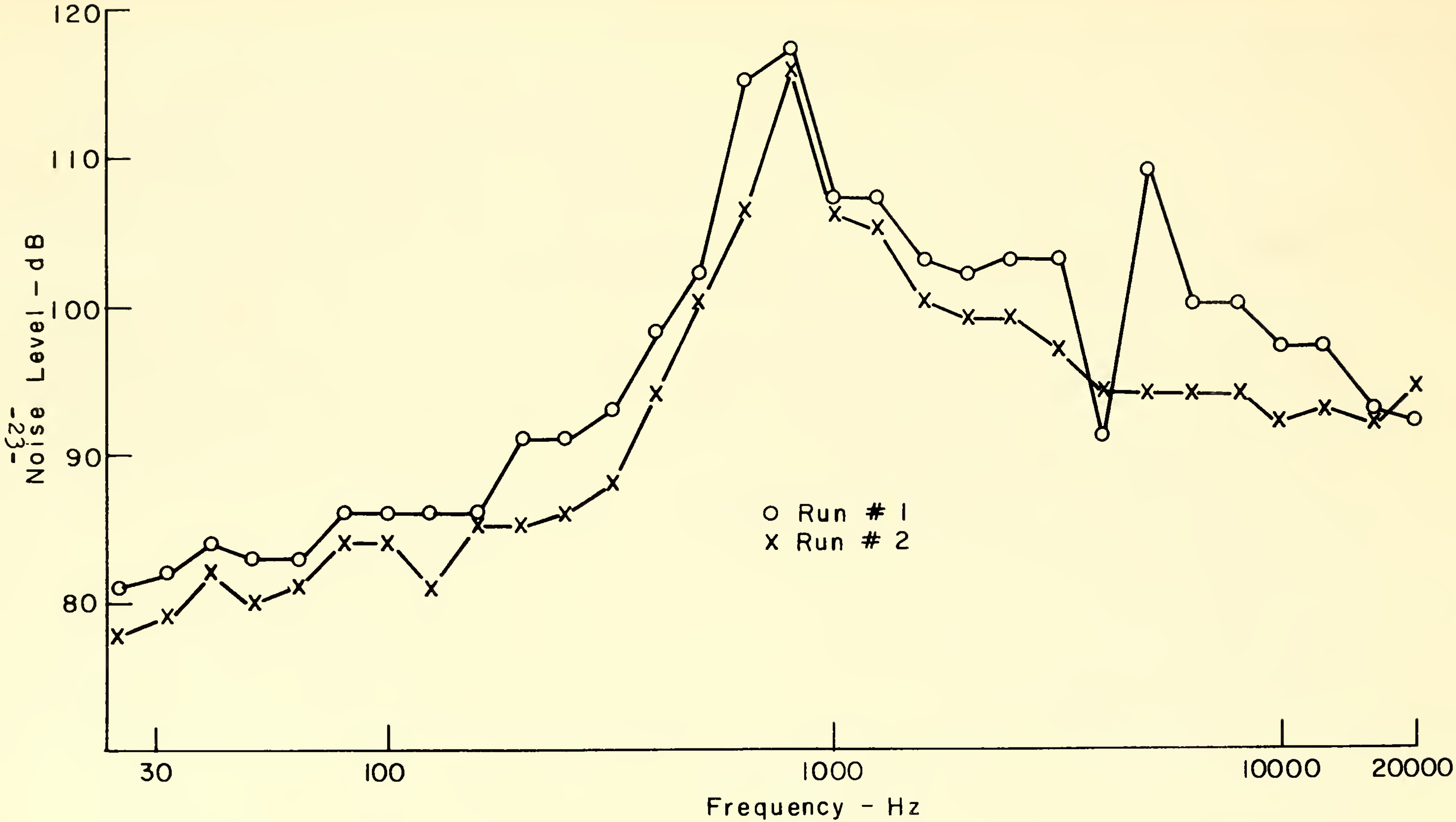


Figure 5—Noise Spectra of Gas Turbine Lab Model.

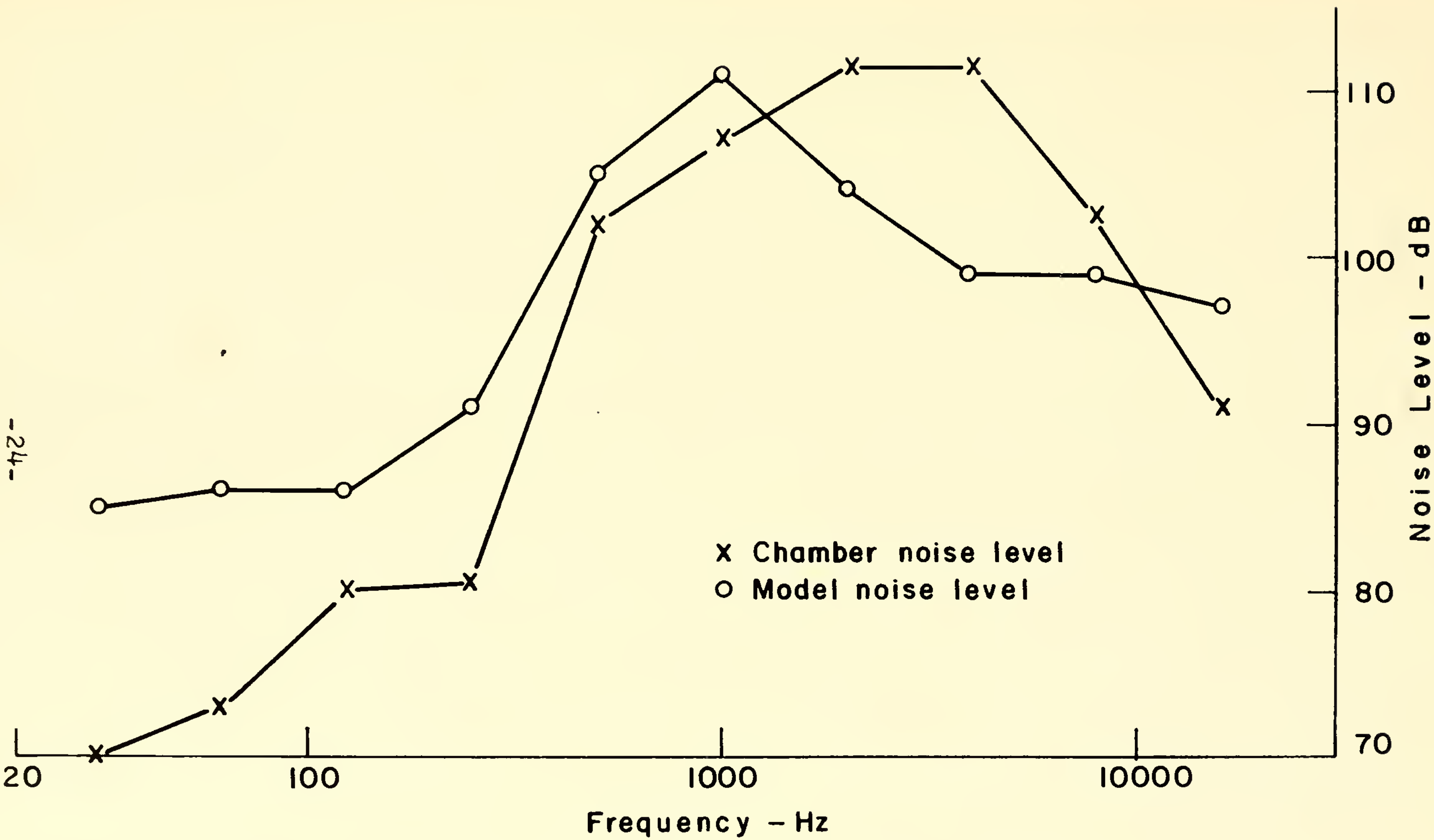


Figure 6 – Comparison of Model and Chamber Noise Levels.

that followed them are shown in Appendix B. The model noise spectrum has a large increase in level in the third octave between 500 and 630 Hz. The noise level drops again above 1000 Hz. A muffler design should emphasize the suppression of noise that falls between 500 and 1000 Hz. The design of a muffler for this particular system is discussed in the next chapter.

CHAPTER IV

A MUFFLER DESIGN

A muffler may be defined as a special duct or pipe that reduces the transmission of sound while permitting the free flow of air. A successful muffler design for hyperbaric chambers must satisfy the following criteria:

1. The acoustical criterion, as established by the Walsh-Healy Act of 1969. That is, that during ventilation of the chamber the sound level shall not exceed 90 dB as determined by a sound level meter operating on the A weighting network with slow meter response. This acoustical criterion would allow continuous ventilation of the chamber.
2. The geometrical criterion which specifies that the muffler must be able to fit under the deck plates of the hyperbaric chambers found at NSDS. This space is approximately 8 inches by 8 inches by 16 inches.
3. The aerodynamic criterion that the pressure drop through the muffler does not restrict the flow rate to the point that the chamber may no longer be pressurized at the rate of 60 feet per minute.
4. The oil-free criterion which specifies that the muffler must not act as a trap for oil which may be in the incoming air. Since the atmosphere in the chamber is at times

heavily laden with oxygen, such oil would act as a fire hazard.

Mufflers are divided into two categories, dissipative and reactive. Dissipative mufflers are those whose acoustical performance is determined predominantly by the presence of flow-resistive material. Reactive mufflers provide an impedance mismatch for the acoustic energy traveling along the duct. This impedance mismatch results in a reflection of part of the acoustic energy. The oil-free criterion restricts the hyperbaric chamber design to one without flow-resistive material that might collect oil.

The acoustical behavior of a muffler can be expressed in terms of the insertion loss (IL). Insertion loss is generally defined as the difference, in decibels, between two sound-pressure levels which are measured at the same point in space before and after a muffler is inserted between the measurement point and the noise source.

Because the fundamental equations of motion governing linear acoustics are formally very similar to the equations of electrical theory, it is possible to study acoustical systems with electrical analogs. Beranek developed an impedance-type of analogy in Reference 9. In his analogy, the quantity that flows through the acoustical elements is

the volume velocity U in cubic meters per second and the drop across the acoustical elements is the pressure p in newtons per square meter. The law of conservation of mass ensures that continuity of volume velocity must exist at a junction of acoustical elements just as in electricity there is continuity of electrical current at a junction. The acoustical pressure p and the volume velocity U are the AC components of the flow through the muffler and should not be confused with the steady air flow. The acoustical elements and their electrical analogs are listed in Table 3.

The first muffler design is shown in Figure 7. It consists of a number of reactive elements which collectively act as an acoustical filter. The air enters the muffler through a 90° elbow at the threaded portion of the central pipe. The air passes through the pipe and past the holes drilled in the end of the pipe. These holes and the chamber surrounding them act both as a pipe-resonator and as a volume-resonator muffler. The air turns in the bottom chamber and exits through the lower baffle plate which contains three one inch diameter holes. This plate and the similar one above it act, together with the intervening volume, as an expanded cross-section muffler (15). The air exits through the outlet plenum after undergoing another 90° turn. The muffler is designed to be "tuned" by changing the length of the spacers surrounding the

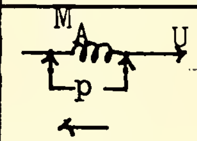
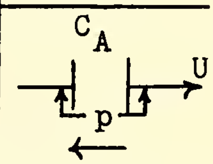
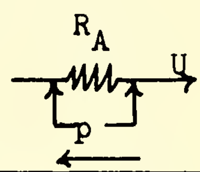
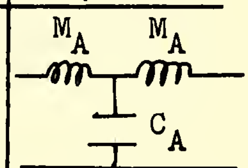
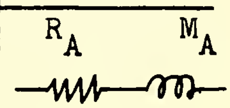
ACOUSTIC ELEMENT	ELECTRICAL VALUE	ACOUSTICAL IMPEDANCE ANALOGY	CIRCUIT SYMBOLS
Pressure	e	p	----
Volume Velocity	i	U	----
Tube of Pipe	L	Mass Element-M_A	
Enclosed Volume of Air	C	Compliant Element-C_A	
Fine-mesh Screen	R	Dissipative Element-R_A	
Cavity with Holes on Opposite Sides		Mixed Mass-Compliance Element	
Intermediate-sized Tube		Mixed Mass-Resistance Element $Z_A=R_A+jwM_A$	

Table 3-Acoustical Elements and Their Electrical Analogs

(Reference 9)

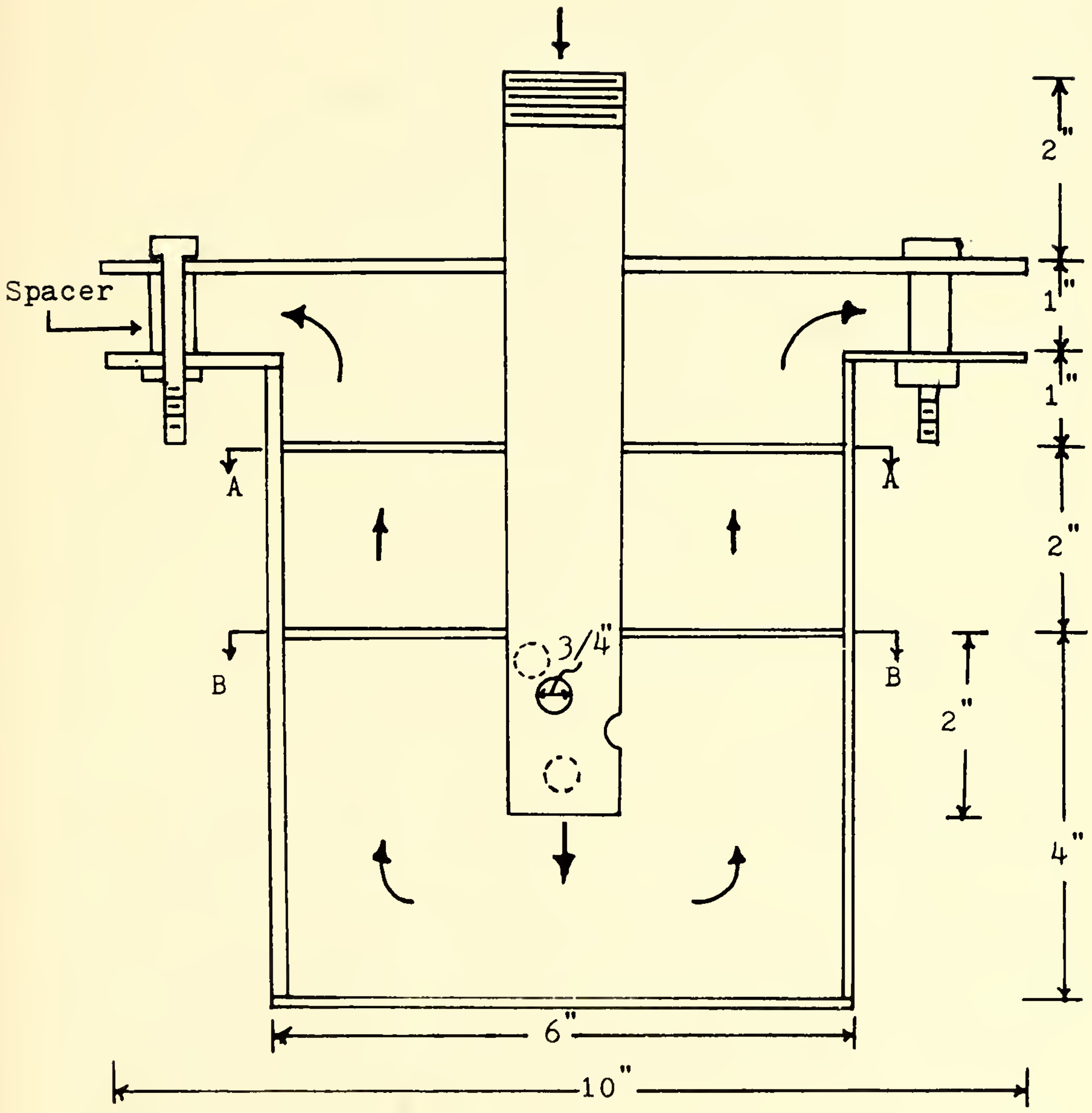


Figure 7-Muffler Design for Hyperbaric Chambers. Sections A-A and B-B are baffle plates, each containing three 1" diameter holes. The heavy arrows indicate air flow.

bolts. The length of the spacers determines the width of the outlet plenum.

The electrical-acoustical analogy implies several simplifying assumptions of which the more important are listed below.

1. The analogy assumes that the impedances may be treated as being "lumped" impedances rather than "distributed". This assumption is valid when the largest cross-sectional dimension is much smaller than a wave length. At 250 Hz the smallest wavelength/diameter ratio is that of the six inch bottom chamber and is about equal to 9.0.

2. The analogy does not consider the effects of flow through the muffler.

3. The temperature variations in the system have been assumed to not affect the sound propagation. The effect of the average temperature on the velocity of sound and wavelength has been taken into account.

4. It has been assumed that sound pressures are small compared with absolute pressure, so that nonlinear effects are negligible.

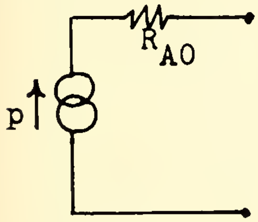
5. It has been assumed that sound is propagated in plane waves, unattenuated by viscosity or heat conduction.

6. It has been assumed that muffler wall surfaces do not conduct or transmit sound.

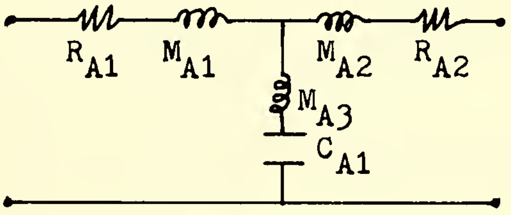
For this application the assumption that flow noise may be neglected appears to be the most serious item on

the list.

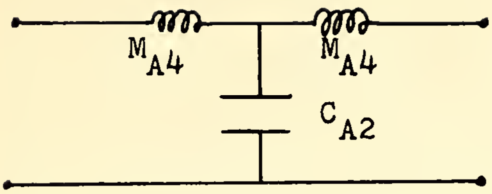
Using the analogous elements in Table 3 the electrical-acoustical analogy for the muffler design has been developed. The piping up to the holes in the muffler is long enough to consider the input impedance to be that of an infinitely long pipe of ID $1\frac{1}{4}$ inches. The holes in the pipe act in union with the lower cavity as a resonator. The pipe exit, lower cavity and lower baffle plate are treated as a mixed-mass compliance element. The cavity with baffle plates above and below it acts as a mixed mass-compliance element. the length of the chamber above the last baffle plate and the outlet flange length are treated as acoustic masses. The termination impedance is that of an unflanged tube. The source of sound is modelled as a constant-pressure generator that is unchanged as a result of inserting the muffler.



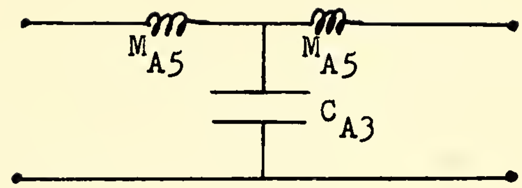
Pressure generator and input impedance



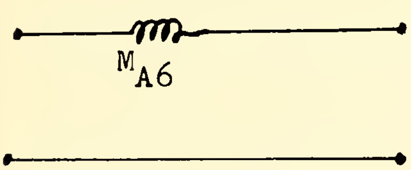
Length of pipe at holes Resonator effect of holes in pipe



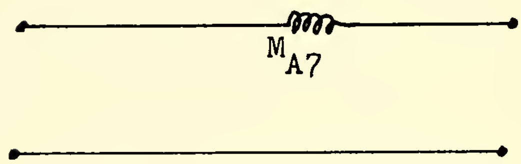
Bottom cavity mixed mass-compliance element



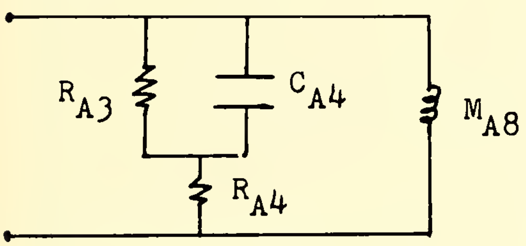
Middle chamber mixed mass-compliance element



Length of chamber above last baffle plate



Outlet flange length



Termination impedance

The element sizes were determined using the following constants:

$P_o = 10^5$ newton/meter^2	$c = 345$ m/second
$\gamma = 1.4$	$a_1 = 0.0162$ m
$\mu = 1.5\text{x}10^{-5}$ m^2/second	$a_2 = 0.0194$ m
$\rho_o = 1.18$ KG/m^3	$a_3 = 0.08$ m

The element sizes were calculated as shown below:

$$R_{A0}=\frac{\rho_0 c}{\pi a_1^2} = 49.4\text{x}10^4 \text{ mks ohms}$$

$$R_{A1}= \frac{1}{\pi a_1^2}\rho_0(2w\mu)^{\frac{1}{2}}\left(\frac{l'}{a_1}\right) = 488 \qquad (\text{Note: } w=2\pi(250))$$

$$M_{A1}= \frac{l'\rho_0}{\pi a_1^2} = 36.4 \text{ KG/m}^4$$

$$M_{A2}=\frac{\rho_0(l'+l'')}{\pi a_1^2} = 50.7$$

$$R_{A2}=\frac{\rho_0(2w\mu)^{\frac{1}{2}}}{\pi a_1^2}\left(\frac{l'}{a_1} + 1\right) = 800$$

$$C_{A1}=\frac{V}{\gamma P_o} = 1.4\text{x}10^{-8} \text{ m}^5/\text{newton}$$

$$M_{A3}=\frac{\rho_0 l'}{4 S_{hole}} = 16.95$$

$$M_{A4}=\frac{\rho_0 l/2}{\pi a_2^2} = 75.9$$

$$C_{A2}= \frac{V}{\gamma P_o} = 1.255\text{x}10^{-8}$$

$M_{A5}= 59.2$

$C_{A3}= 0.206\text{x}10^{-8}$

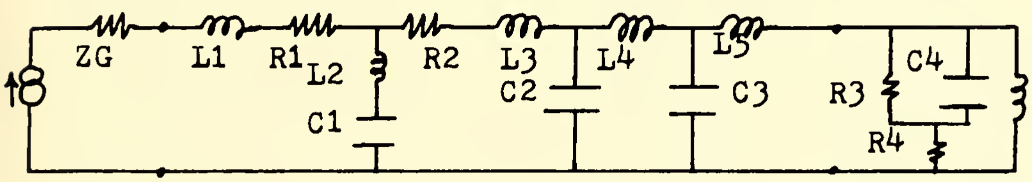
$M_{A6}= 1.61$

$M_{A7}= 1.93$

Termination Impedance

	One Inch Spacers	One-half Inch Spacers	$1\frac{1}{2}$ Inch Spacers
$R_{A3} = 0.1604 \frac{\rho_o c}{a_3^2} =$	10^4	$2x10^4$	$6.7x10^3$
$R_{A4} = 0.318 \frac{\rho_o c}{a_3^2} =$	$2x10^4$	$4x10^4$	$1.3x10^4$
$C_{A4} = \frac{5.44}{\rho_o c^2} a_3^3 =$	$2x10^{-8}$	$7x10^{-9}$	$3.7x10^{-8}$
$M_{A8} = 0.1952 \frac{\rho_o}{a_3} =$	2.85	4.0	2.3

The analogy was simplified as shown below.



GENERATOR NETWORK LOAD

Where:			
	$ZG=R_{A0}$	$R2=R_{A2}$	$L5=M_{A5}+M_{A6}+M_{A7}$
	$R1=R_{A1}$	$C2=C_{A2}$	$R3=R_{A3}$
	$L1=M_{A1}$	$L3=M_{A2}+M_{A4}$	$R4=R_{A4}$
	$L2=M_{A2}$	$L4=M_{A4}+M_{A5}$	$C4=C_{A4}$
	$C1=C_{A1}$	$C3=C_{A3}$	$L6=M_{A8}$

This muffler design was analyzed using a computer program named MARTHA. MARTHA was developed by Professor Paul Penfield, Jr. at the Massachusetts Institute of Technology (10). MARTHA is designed to analyze linear electrical networks. MARTHA is geared toward "transmission-type" networks such as this one with an input and an output. One of the many response functions generated by MARTHA is the transducer gain TG which is the ratio of P_{out} to P_{ga}, where:

$$P_{out} = \frac{|E_L|^2}{4\ \mathrm{Re}\ (ZL)}$$

E_L is the output voltage with the termination impedance ZL in place.

$$P_{ga} = \frac{|E_g|^2}{4ZG}$$

E_g is the input voltage with the generator load ZG in place.

$$TG = \frac{P_{out}}{P_{ga}}$$

Expressed in decibels, this becomes DB TG

Insertion Loss of Muffler =IL= DB TG

The insertion loss IL is shown plotted in Figure 8 for a muffler with one inch spacers. The computer input and other plots of IL are given in Appendix C.

CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   10:12
MUFFLERS WITH ONE INCH SPACERS-DESIGN ONE

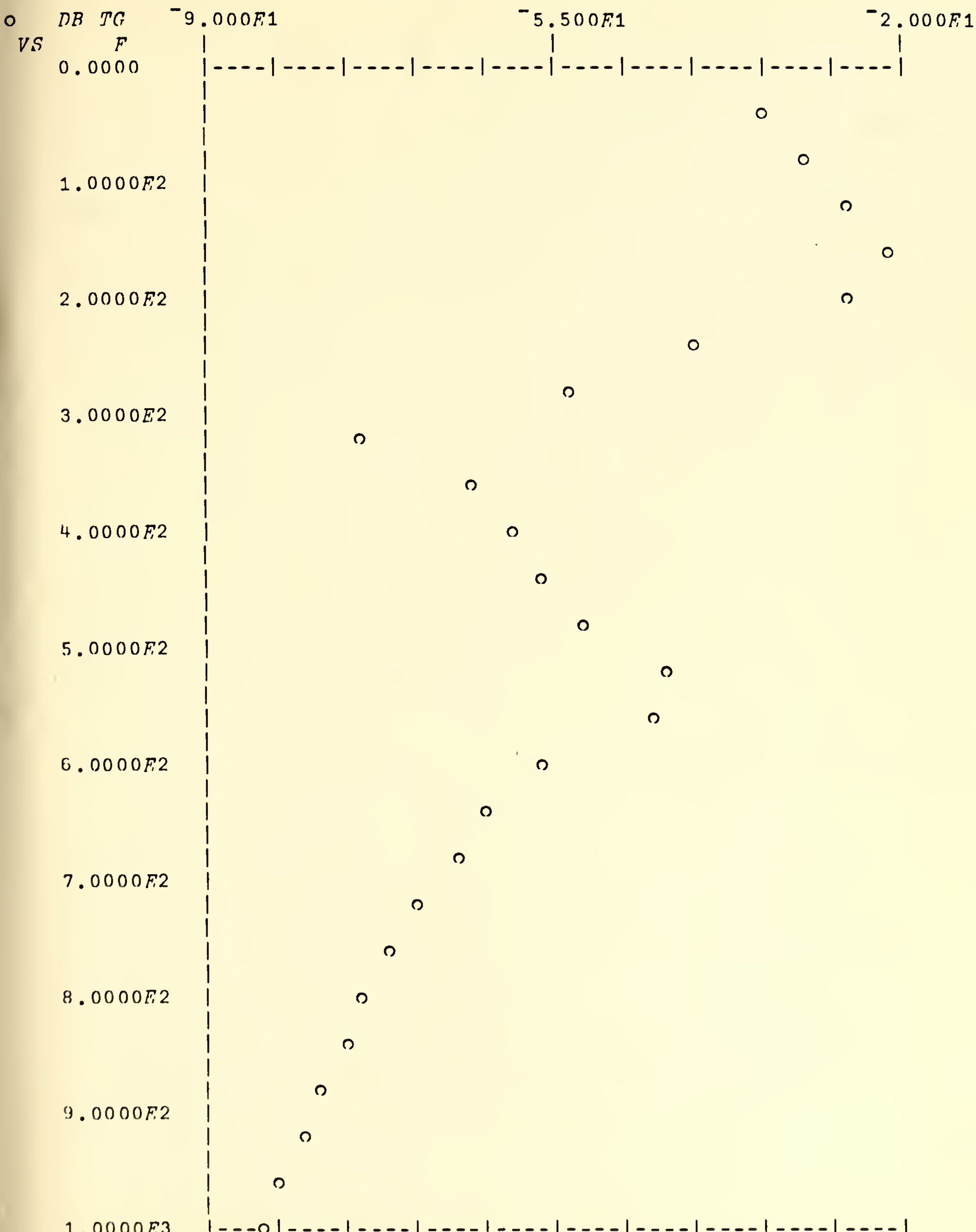


Figure 8-Plot of IL as Predicted by the Analogy

CHAPTER V

TEST AND EVALUATION OF THE MUFFLER DESIGN

The final stage of the design of the muffler consists of experimental noise measurements with the muffler in place. The muffler was installed on the air pipe as shown in the photograph (Figure 9). Runs two and three were with the throttling valve $\frac{1}{4}$ turn open (about five turns is required to open the valve fully). In run two with no muffler, the flow rate was 102 SCFM. In run three with the muffler the flow rate was 77 SCFM. Full details of the runs may be found in Appendix B.

The noise spectra found in these two runs is plotted on Figure 10. The A-weighted sound level was reduced from 115 dB to 106 dB as a result of inserting the muffler. The insertion loss at the peak noise level was 13 dB at 800 Hz. Above 2000 Hz the muffler performed quite well, reducing the noise level by as much as 20 dB. The measured insertion loss obtained fell short of that predicted by the electrical-acoustical analogy. The average measured insertion loss for runs 1-4 is compared to the predicted insertion loss on Figure 11. The measured insertion loss bears little relationship to the expected insertion loss. This result could come about if the assumptions underlying the analog are invalid or if the analog did not accurately describe the muffler.

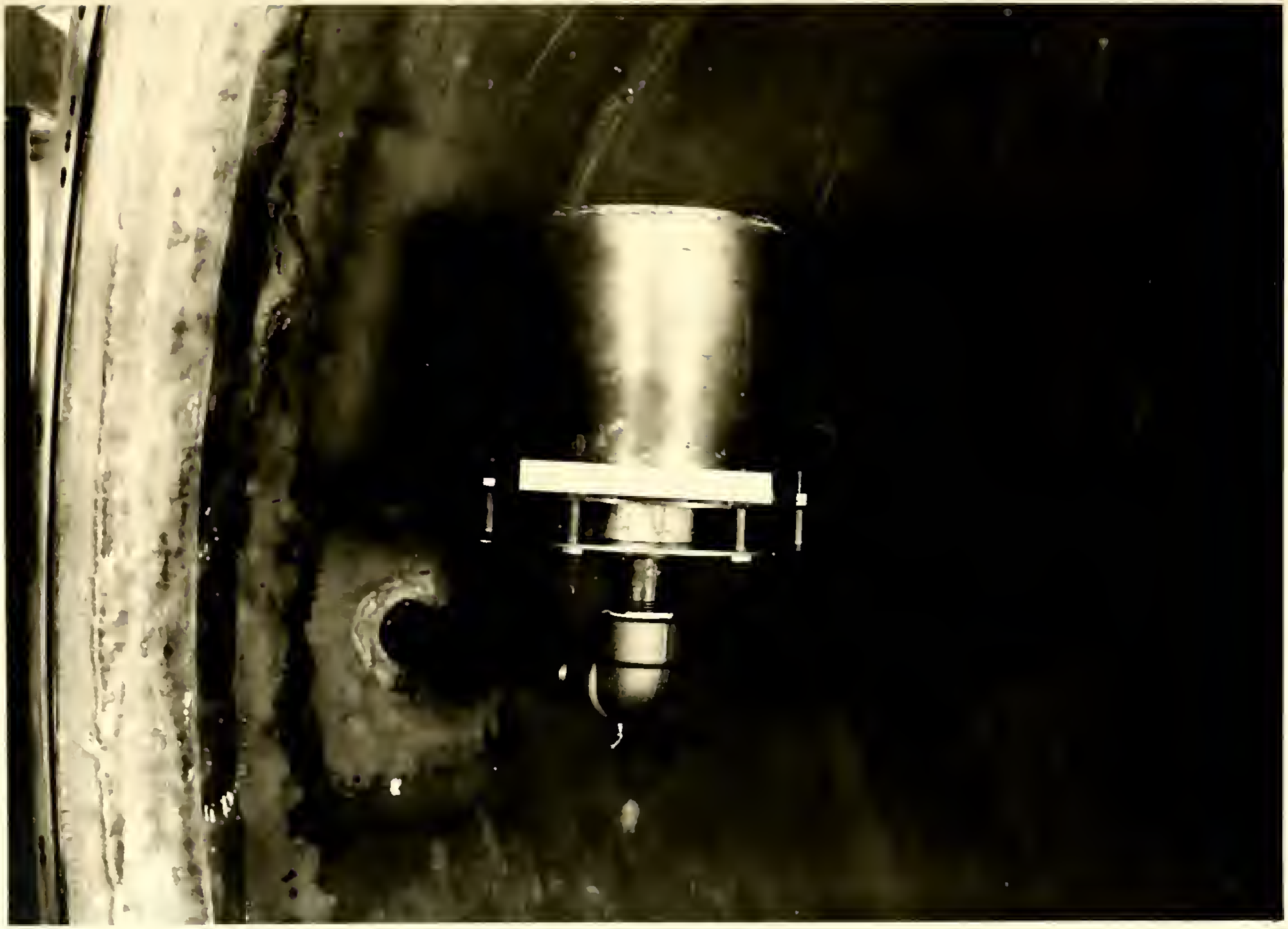


Figure 9-Muffler Installed in Gas Turbine Lab Model. Note six-inch rule.

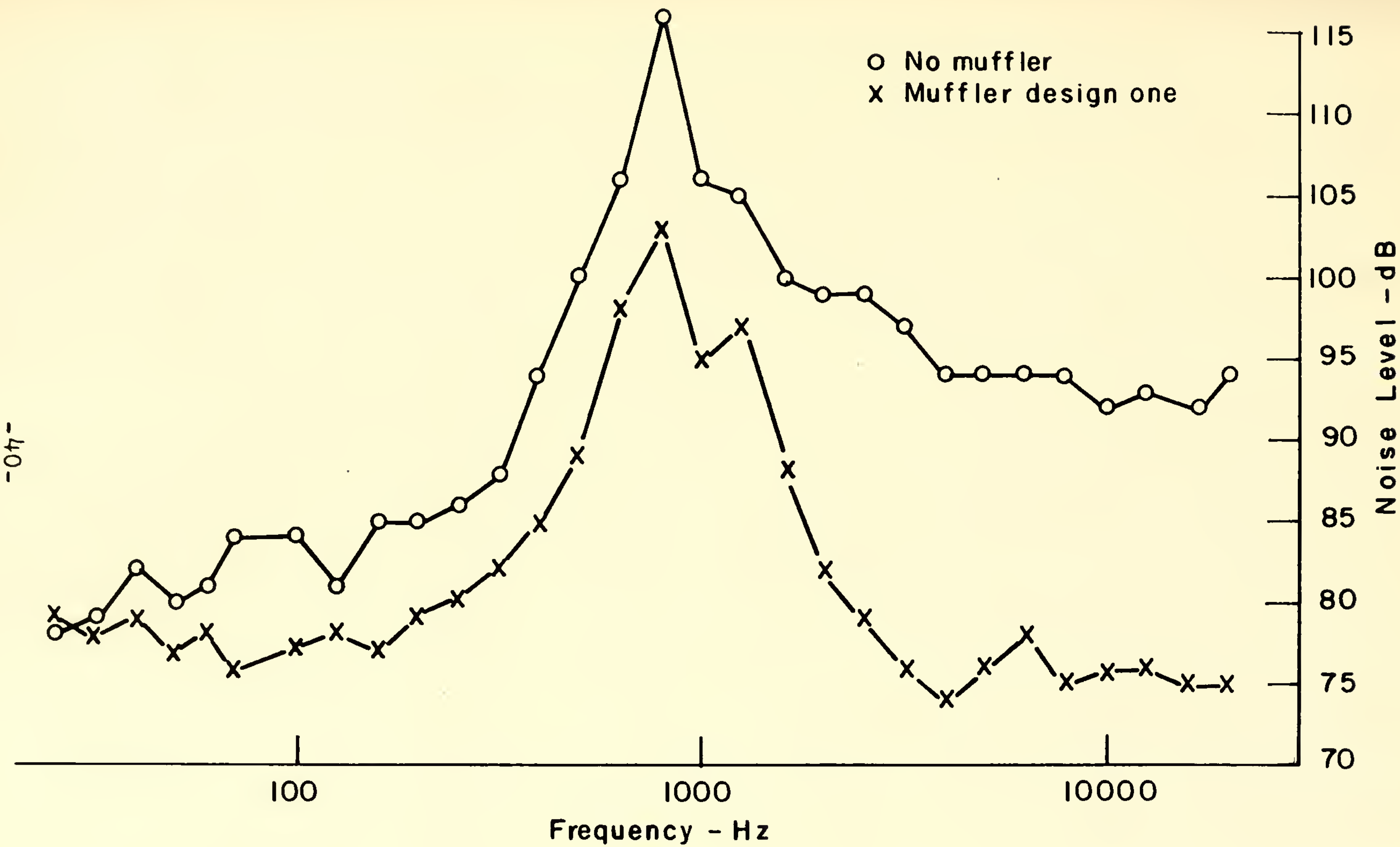


Figure 10—Noise Spectra of the Gas Turbine Lab Model without any muffler (run 2) and with muffler design one (run 3).

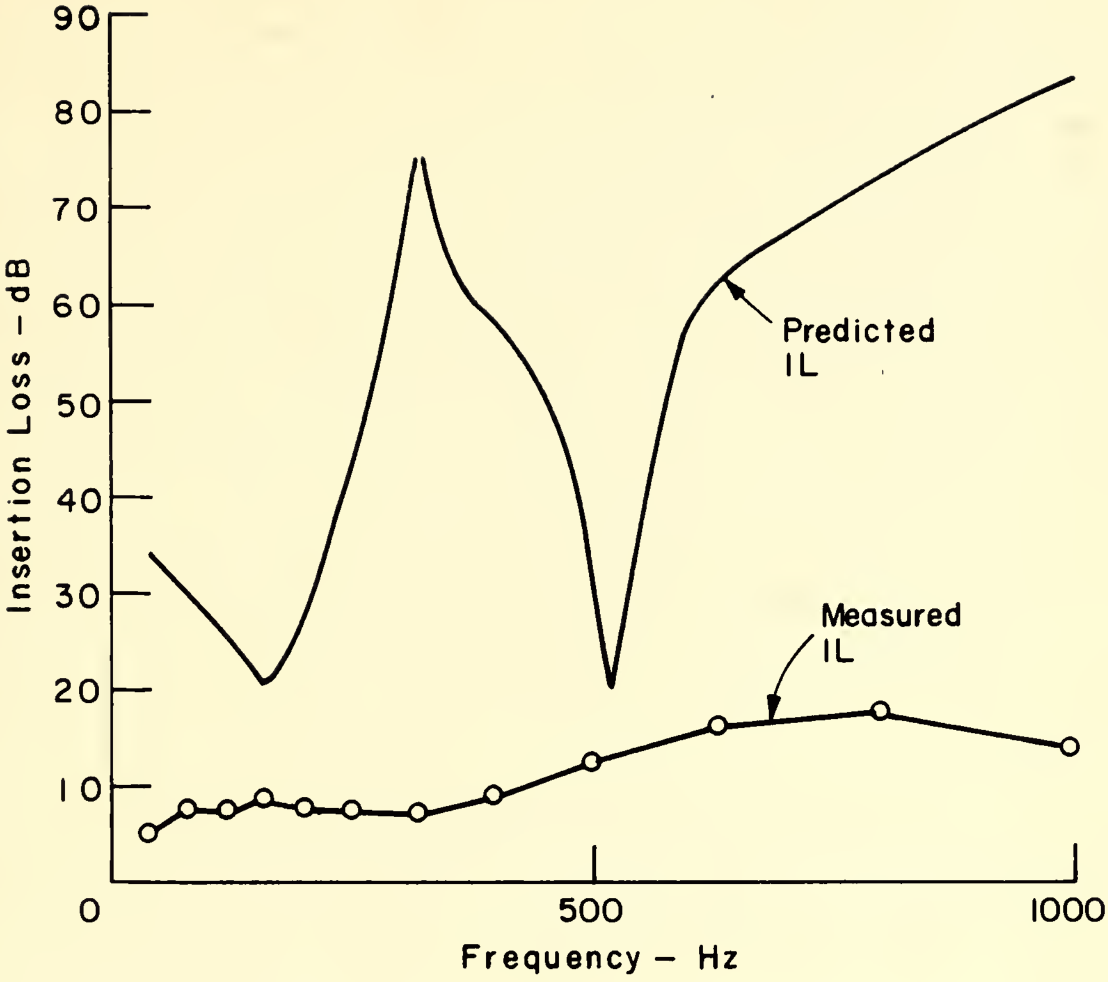


Figure 11- A comparison of the measured and the predicted IL resulting from inserting muffler design one.

The rush of air through the muffler does cause self noise to be generated and it could cause the characteristics of the various acoustical elements to change.

At a flow rate of 77 SCFM the velocity of the air leaving the $1\frac{1}{4}$ inch pipe would be about 45 m/sec (assuming the air is at nearly atmospheric pressure). This air is entering the portion of the muffler modelled both as a pipe-resonator and a volume resonator.

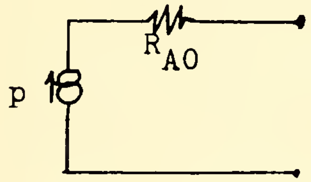
The insertion loss of a pipe-resonator is reduced considerably by a steady flow, with the reduction occurring mainly in the lower frequencies (12). The insertion loss of pipe-resonators is reduced rapidly at air velocities above 15 m/sec.

Volume-resonators suffer a decrease in insertion loss as the flow velocity increases. This effect is strongest near the resonance frequency. For this muffler design the resonance frequency is 330 Hz (13).

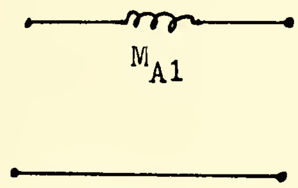
The velocity of the air entering the lower baffle plate is about 25 m/sec. The insertion loss of a simple expansion chamber is not affected by the presence of superimposed steady flow up to a velocity of about 35 m/sec (12).

From the above discussion it would seem evident that the steady flow of air has a considerable effect on the acoustical performance of the muffler. Since the pipe-

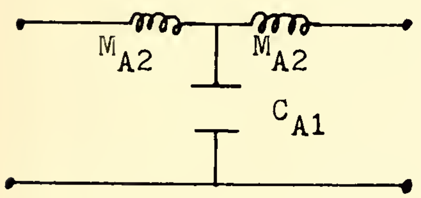
resonator and volume-resonator are so completely masked by the flow effects, an analog model of the muffler substituting an expanded cross-section chamber for the resonators was developed as shown below.



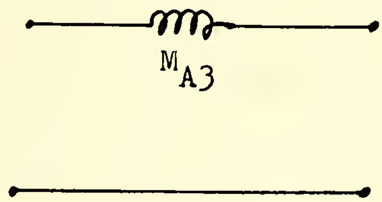
Pressure generator



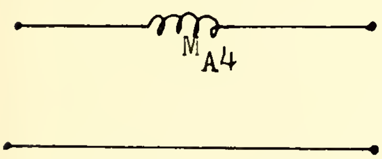
Length of bottom chamber



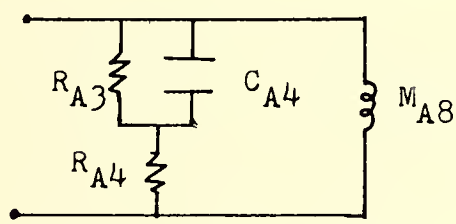
Middle chamber mixed mass-compliance element



Length of chamber above last baffle plate

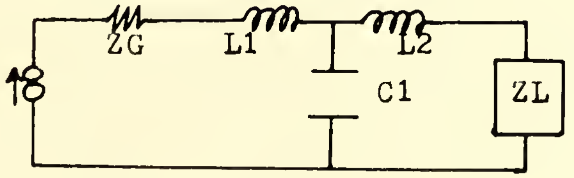


Outlet flange length



Termination Impedance

The simplified network was:



Where: $ZG = R_{AO} = 49{,}400$ mks ohms

$L1 = M_{A1} + M_{A2} = 21\ KG/m^4$

$C1 = C_{A1} = 5.55 \times 10^{-9}\ m^5/\text{newton}$

$L2 = M_{A2} + M_{A3} + M_{A4} = 63\ KG/m^4$

ZL = Same as before

The revised analog was analyzed using MARTHA as before and the predicted insertion loss is shown in Figure 12.

The termination impedance ZL has been defined as that of an unflanged tube with no air flow. The air flow causes the air surrounding the outlet plenum to move. This air that is accelerated without compression represents an increased acoustic impedance. Figure 13 compares the predicted and average measured insertion loss. Two values of L6 are used, one for the no flow case ($L6=2.85\ KG/m^4$) and one for a value of $L6=14.3\ KG/m^4$.

The predicted insertion loss is shown to be much more like the measured insertion loss for the revised analog than for the original analog. Once again it must be emphasized that the self-noise generated by the air flow is not considered in the electrical-acoustical analogy.

The muffler design was analyzed using the various length spacers. The analysis indicated little effect on the insertion loss as the spacer length was changed. These results may be seen in detail in Appendix C.

The muffler design was modified to allow air flow to

CIRCUIT ANALYSIS BY MARTHA. 73∘A 4/27/73 10:24
MUFFLER WITH ONE INCH SPACERS-REVISED ANALOGY

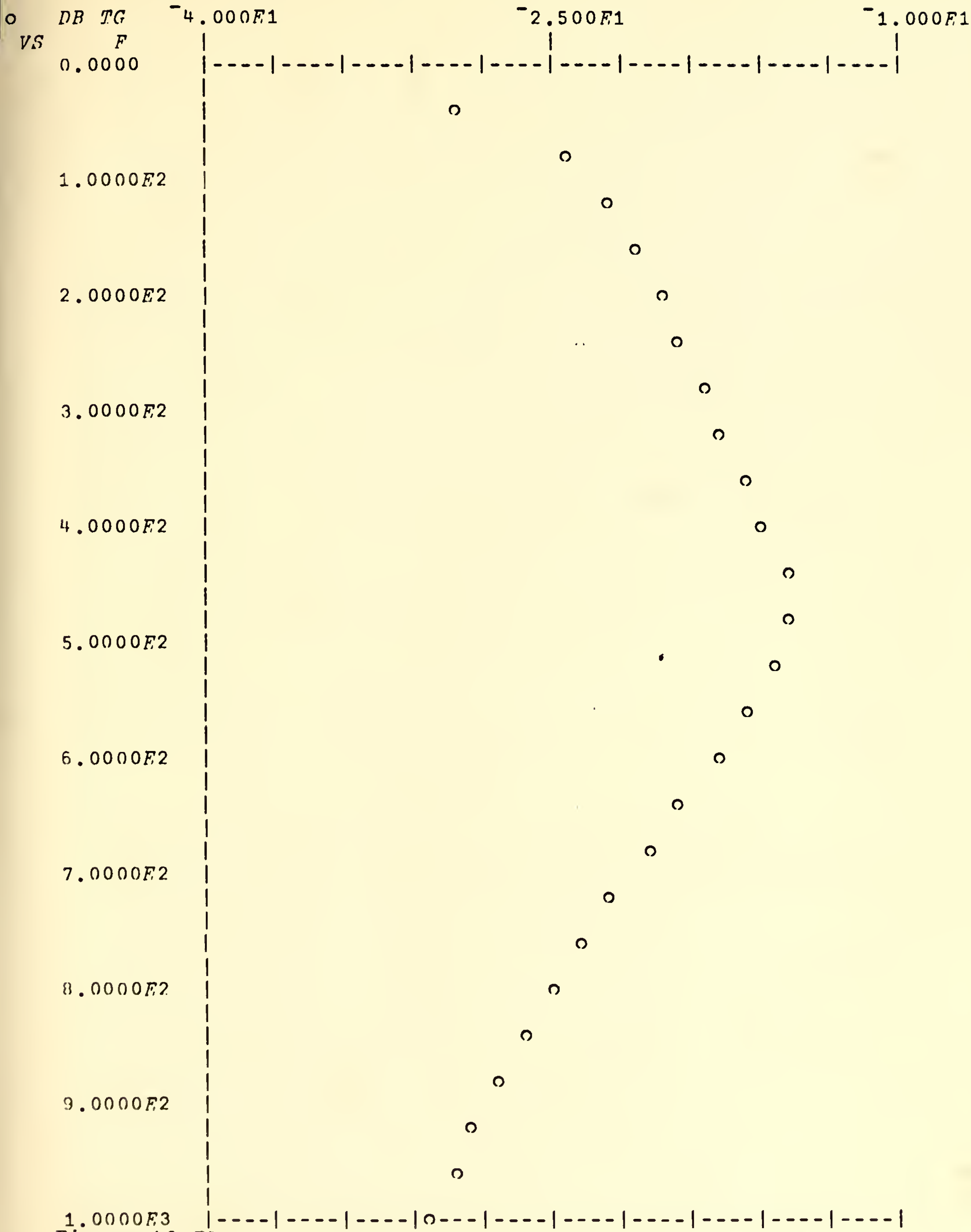


Figure 12-Plot of IL as Predicted by the Revised Analogy.

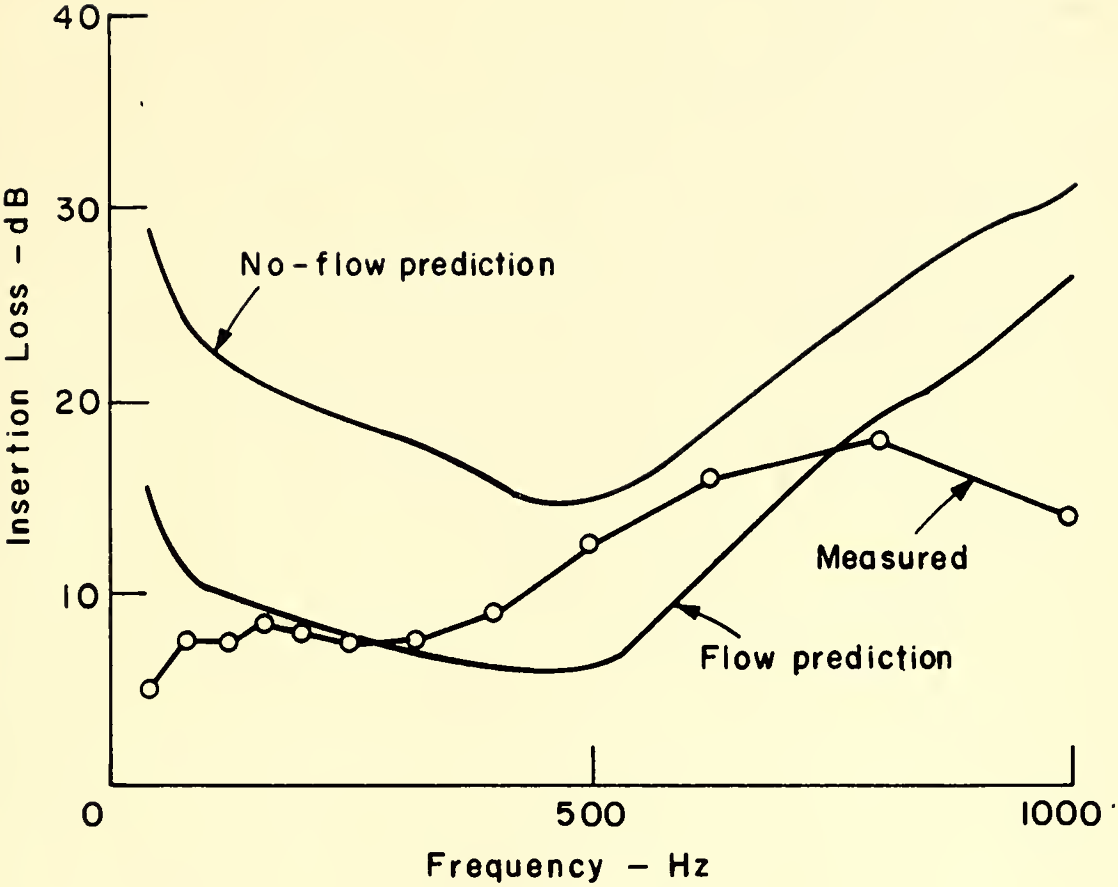


Figure 13—A comparison of the measured and predicted IL resulting from muffler design one. The predicted IL is from the revised analogy.

bypass the bottom chamber with the intention of having the bottom chamber act as a resonator as it was originally intended. Runs five and six were made with four one-half inch ID holes drilled in the inlet pipe. These four holes allowed air to enter the middle chamber without passing through the bottom chamber. The results of this modification were as shown in Figure 14. The A-weighted sound level was 112 dB for run five and 109 dB for run six. This compares with 106 dB for the original muffler design (run three).

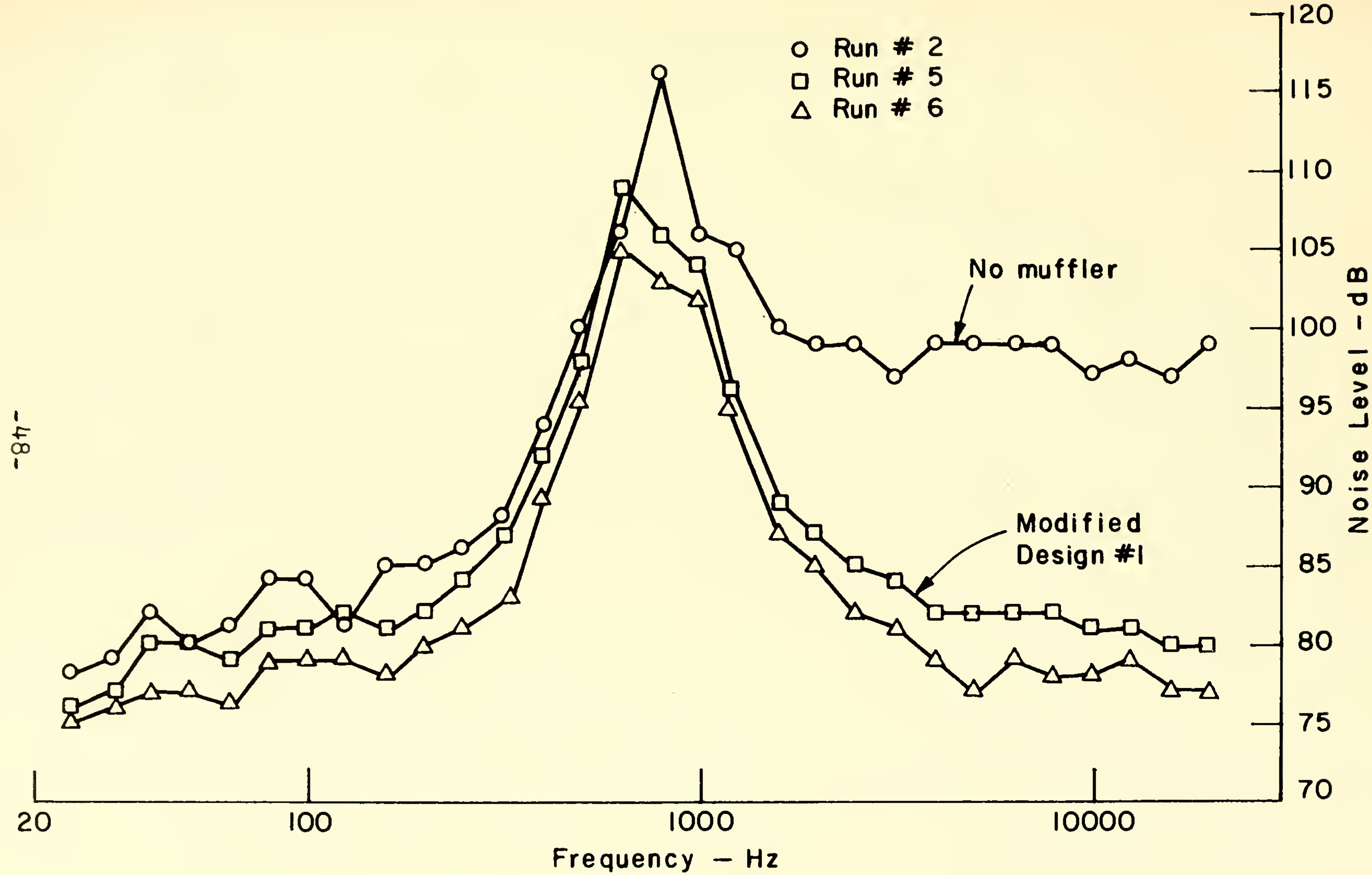


Figure 14—Noise Spectra of the Gas Turbine Lab Model without any muffler (run 2) and with modified muffler design one (runs five and six) under similar conditions.

CHAPTER VI

A SECOND MUFFLER DESIGN

Inserting the first muffler design into the system did not allow the system to meet the acoustical criterion as established in Chapter IV. In an attempt to improve the insertion loss, a second muffler was designed and built.

The first muffler design resulted in an insertion loss that was difficult to predict. At higher frequencies (500 to 1000 Hz) the insertion loss indicated that the first muffler was acting as an expanded cross-section chamber. The ability to predict the insertion loss was hampered by self-noise generated in the muffler and an altering of the characteristics of the acoustical elements.

The second muffler design was designed to reduce the effect of air flow on the self-noise generated by reducing the velocity of the incoming air as early as possible. The slower-moving air is then passed through three expanded cross-section chambers, the acoustical component whose characteristics are least effected by a steady air flow. The selection of the dimensions of the three expanded cross-section chambers is based on methods described in Reference 12. The behavior of a chamber can be described in terms of two parameters, m and kl, where:

$$m = \frac{\text{cross-sectional area of chamber}}{\text{cross-sectional area of duct}}$$

$kl = \frac{2\pi l}{\lambda}$ $\quad$ l=length of chamber

λ =wavelength of sound at the temperature of the gas in the chamber

The transmission loss (TL) of the chamber in the absence of a steady air flow is given by:

$$TL = 10 \log(1+\tfrac{1}{4}(m-\tfrac{1}{m})^2 \sin^2 kl) \quad dB$$

Transmission loss of a muffler is defined as 10 times the logarithm, to base 10, of the ratio of sound power incident on the muffler to the sound power transmitted by the muffler. In order for the above expression to be valid, the inlet and outlet tubes must be infinitely long, or themselves contain mufflers with impedance equal to $\rho_0 c$. The method is, therefore, valid only for each chamber individually and will not predict the transmission loss from the three chambers in series. The expanded cross-section chambers are designed to maximize the transmission loss around center frequencies of 500 Hz, 800 Hz, and 1250 Hz since the noise spectra of the air system indicates that at these frequencies the noise is greatest.

The second muffler design is shown in Figure 15. The same external dimensions are used as in the first design.

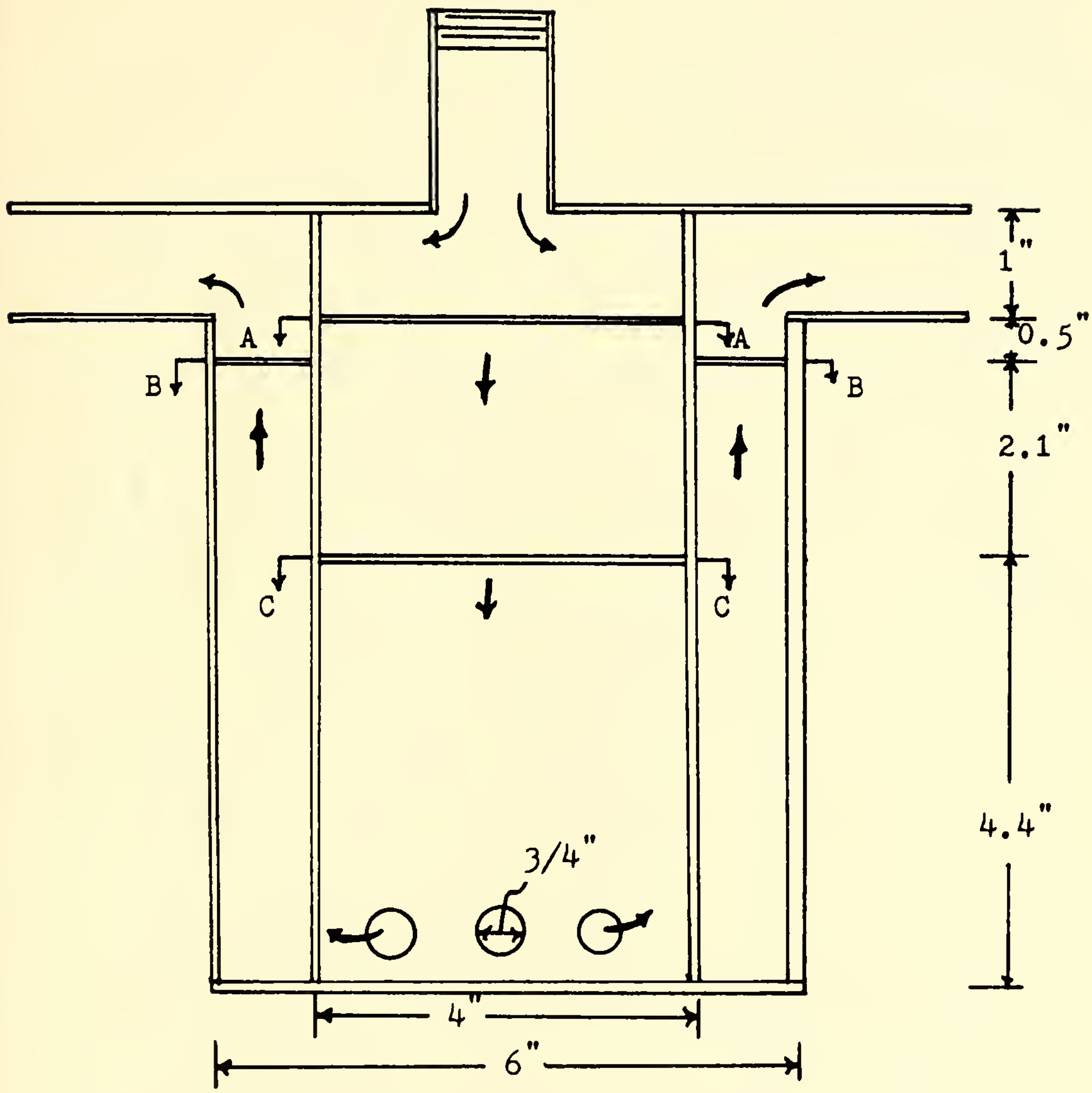
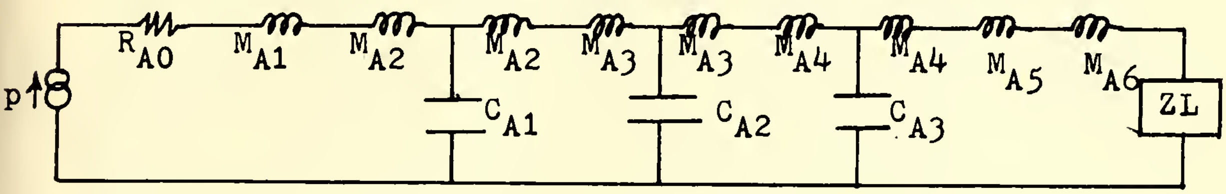


Figure 15-A Second Muffler Design for Hyperbaric Chambers. Sections A-A, B-B, and C-C are baffle plates with holes for air flow. The heavy arrows indicate air flow.

The air enters the muffler through the $1\frac{1}{4}$ inch ID pipe. It enters a 4 inch diameter chamber where the air velocity is reduced. At 140 SCFM the air velocity through the muffler will not exceed 35 m/sec after the air leaves the initial chamber. The air then flows through four 1 inch holes in baffle plates A-A and C-C. The air will turn and flow through seven 3/4 inch holes in the bottom of the chamber. The air will flow through seven 3/4 inch holes in baffle plate B-B and out through the outlet plenum.

The electrical-acoustical analog of the muffler was developed as shown below.



Where: Input Impedance; $R_{A0} = 49{,}400$ mks ohms

Inlet Plenum; $M_{A1} = \frac{l\rho_o}{\pi a_4^2} = 3.69\ \text{KG/m}^4$

First Chamber; $M_{A2} = \frac{l\rho_o}{\pi a_5^2} = 12.6\ \text{KG/m}^4$

$C_{A1} = \frac{V}{\gamma P_o} = 0.29\times10^{-8}\ \frac{m^5}{\text{newton}}$

Second Chamber; $M_{A3} = \frac{l\rho_o}{\pi a_6^2} = 12.6\ \text{KG/m}^4$

$C_{A2} = \frac{V}{\gamma P_o} = 0.44\times10^{-8}\ \frac{m^5}{\text{newton}}$

Third Chamber: $M_{A4} = \frac{1\rho_o}{\pi a_7^2} = 1.48\ KG/m^4$

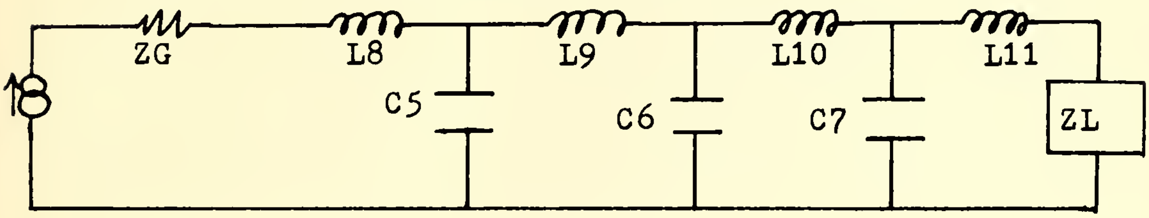
$$C_{A3} = \frac{V}{\gamma P_o} = 0.9\times10^{-8}\ \frac{m^5}{newton}$$

Outlet Chamber: $M_{A5} = \frac{1\rho_o}{\pi a_8^2} = 1.48\ KG/m^4$

Outlet Plenum: $M_{A6} = \frac{1\rho_o}{\pi a_3^2} = 1.93\ KG/m^4$

The termination impedance is assumed to be the same as in the previous analogs. This design does not allow for any but the one inch spacers to be used.

The analogy was simplified as shown below.



Where: $ZG = R_{A0}$

$L8 = M_{A1} + M_{A2}$

$L9 = M_{A2} + M_{A3}$

$L10 = M_{A3} + M_{A4}$

$L11 = M_{A4} + M_{A5} + M_{A6}$

$C5 = C_{A1}$

$C6 = C_{A2}$

$C7 = C_{A3}$

ZL= Same as before

The analog was analyzed using MARTHA as before and the predicted insertion loss is shown in Figure 16. The predicted and the average measured insertion losses are compared in Figure 17.

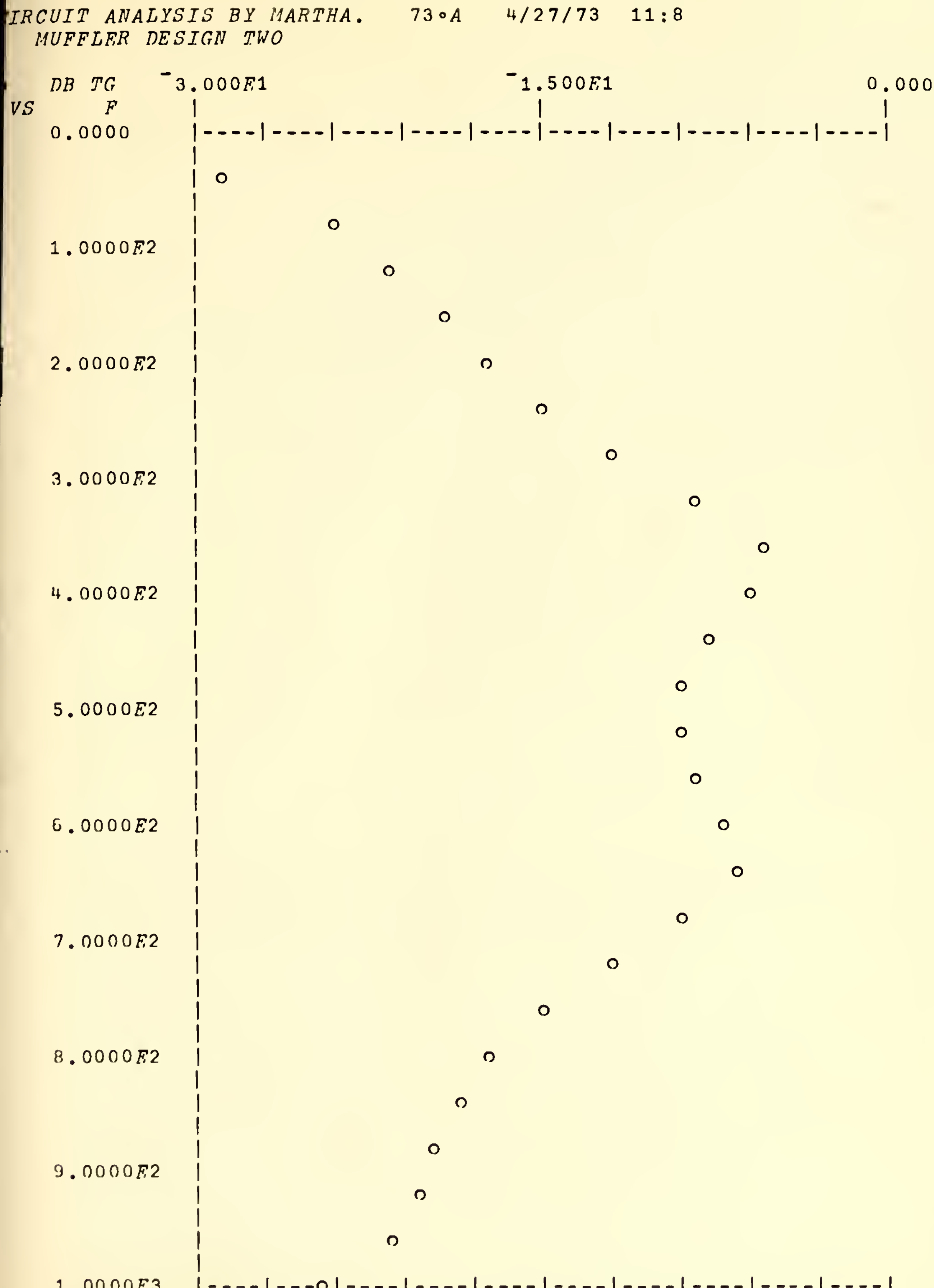


Figure 16-Plot of IL as Predicted by the Analogy.

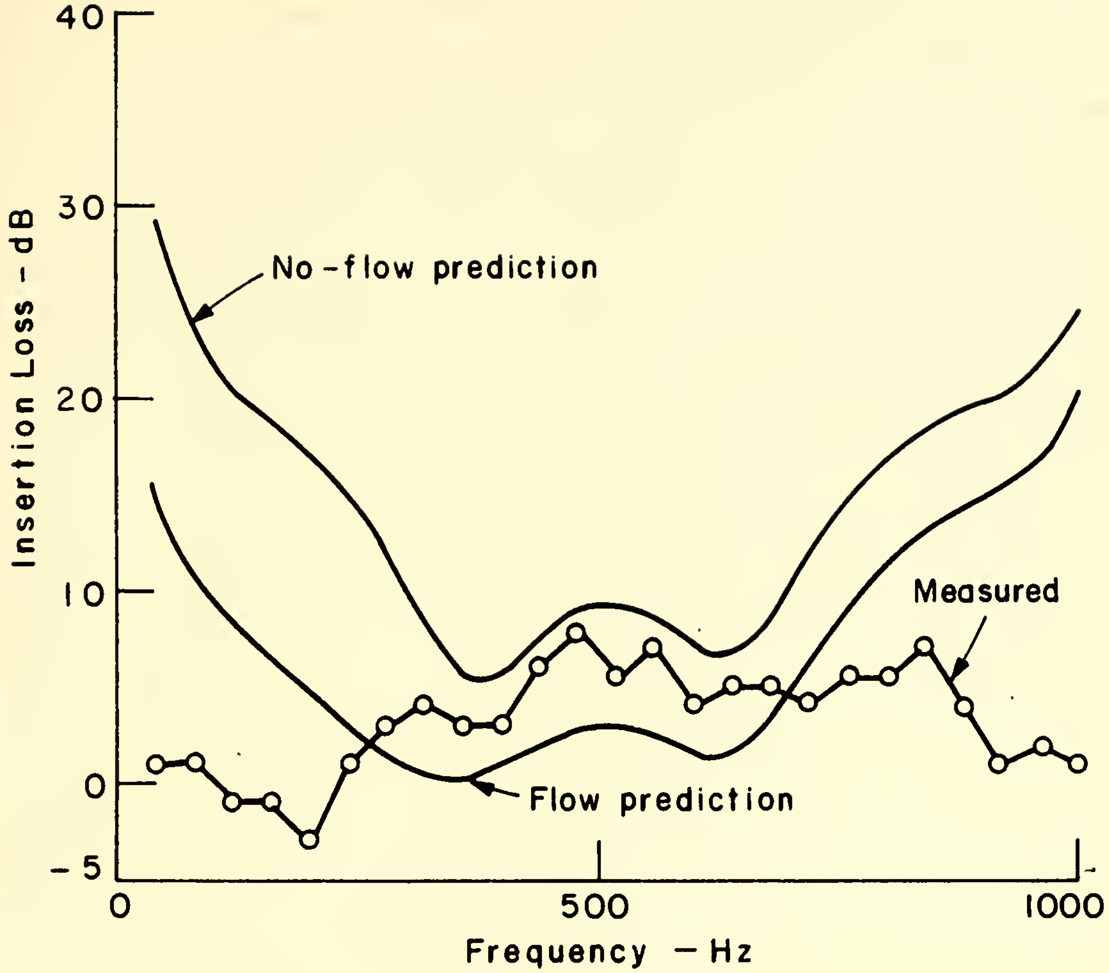


Figure 17 — A comparison of the measured and predicted IL resulting from muffler design two.

The A-weighted sound level was reduced an average of 5 dB as a result of inserting the muffler design two. The air flow rate was 159 SCFM for run seven and 84 SCFM for run ten.

CHAPTER VII

CONCLUSIONS

This thesis demonstrates that it is possible to reduce the air noise found in hyperbaric chambers by inserting a reactive muffler into the air supply system.

The two muffler designs discussed in this paper failed to meet the acoustical criterion as established in Chapter IV. Muffler design one reduced the A-weighted sound level an average of 11 dB. Muffler design two reduced the A-weighted sound level an average of 5 dB. The A-weighted sound level was never reduced to 90 dB.

The geometrical criterion was met by both muffler designs.

The aerodynamic criterion must be established by the flow characteristics of the air system of the particular chamber for which the muffler is designed. Muffler design one reduced the flow rate an average of 32.5 SCFM. Muffler design two had no effect on the flow rate.

The oil-free criterion is met by both muffler designs. Any impurities in the incoming air could be carried through the mufflers by properly designing the baffle plates to direct moisture toward the air stream.

The experimental results from the two muffler designs indicate that a degradation in aerodynamic performance accompanies an increase in insertion loss.

The acceptable aerodynamic performance is determined by the characteristics of the system involved.

The electrical-acoustical analogy is a useful tool in designing mufflers. This tool is limited by the users ability to predict the termination impedance under varying conditions of flow. The electrical-acoustical model loses its ability to predict insertion loss as the frequency increases. In the model, at frequencies above about 700 Hz the acoustical elements no longer can be described as being "lumped" impedances.

CHAPTER VIII

RECOMMENDATIONS

The reduction of air noise in hyperbaric chambers can be affected in a number of ways.

The components in the air supply and exhaust systems generating excessive noise could be replaced with quieter components.

A dissipative or reactive muffler could be inserted into the air supply system downstream of the noise-making components and prior to entering the chamber.

A muffler could be developed that meets the criteria as listed in Chapter IV. Such a muffler would have a large flow area to allow the aerodynamic and acoustical criteria to be met simultaneously.

Since the noise levels are excessively high in hyperbaric pressure chambers, the recommendation is made that all noise reduction methods are investigated.

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APPENDIX A

TOTAL DURATION OF NOISE EXPOSURE ALLOWABLE DURING AN EIGHT-HOUR DAY (Reference 3)

The sound level shall be determined by a sound level meter operating on the A-weighting network with slow meter response. Exposure shall not exceed that shown below:

Duration per Day in Hours	Sound Level dBA
8	90
6	92
4	95
3	97
2	100
1½	102
1	105
½	110
¼ or less	115

These values apply to total time of exposure per working day regardless of whether this is one continuous exposure or a number of short-term exposures but does not apply to impact or impulsive type of noises.

When the daily noise exposure is composed of two or more periods of noise exposure of different levels, their combined effect should be considered, rather than the individual effect of each. If the sum of the following fractions:

$$\frac{C_1}{T_1} + \frac{C_2}{T_2} + \ldots \frac{C_n}{T_n}$$

exceeds unity, then the mixed exposure should be considered to exceed the threshold limit value. C_1 indicates the total time of exposure at a specified noise level, and T_1 indicates the total time of exposure permitted at that level.

APPENDIX B

EXPERIMENTAL DATA

When making a spectrum analysis the Frequency Analyzer Type 2107 switch settings were as required for Frequency Analysis (page 18 of Reference 11) with the following particulars:

1. Weighting Network: "Linear 20-40000 Hz"
2. Frequency Analysis Octave Selectivity: "25 dB"

The switch settings allowed a one third octave linear sound level to be taken at a selected frequency.

When taking the A-weighted sound level the settings were as above with the following exceptions:

1. Weighting Network: "Curve A"
2. Meter Switch: "Slow R.M.S."
3. Function Selector: "Selective Section Off"

The switch settings allowed the sound level meter to operate as required by Appendix A.

The data sheets are reproduced on the following pages. All of the readings were made directly with the exception of the Average Flow Rate which was calculated from the drop in pressure.

DATA

RUN NUMBER	1	2	3	4	5	6
SPACER SIZE(INCHES)	-	-	1	1	1	1
VALVE POSITION(TURNS)	5	$\frac{1}{4}$	$\frac{1}{4}$	5	$\frac{1}{4}$	$\frac{1}{4}$
INITIAL PRESSURE(PSIG)	1920	1580	1340	1910	1780	1500
FINAL PRESSURE(PSIG)	1610	1340	1110	1550	1500	1280
ELAPSED TIME(MIN)	5	5.5	7	8	6.5	6
AVE FLOW RATE(SCFM)	145	102	77	105	100	85
A-WEIGHTED LEVEL(dB)	118	115	106	105	112	109
LINEAR LEVEL-25 Hz (dB)	81	78	79	73	76	75
32	82	79	78	73	77	76
40	84	82	79	77	80	77
50	83	80	77	76	80	77
63	83	81	78	76	79	76
80	86	84	76	79	81	79
100	86	84	77	79	81	79
125	86	81	78	79	82	79
160	86	85	77	77	81	78
200	91	85	79	81	82	80
250	91	86	80	82	84	81
320	93	88	82	84	87	83
400	98	94	85	89	92	89
500	102	100	89	88	98	95
630	115	106	98	90	109	105
800	117	116	103	94	106	103
1000	107	106	95	89	104	102
1250	107	105	97	89	96	95
1600	103	100	88	85	89	87
2000	102	99	82	84	87	85
2500	103	99	79	83	85	82
3200	103	97	76	81	84	81
4000	91	94	74	80	82	79
5000	109	94	76	81	82	77
6300	100	94	78	82	82	79
8000	100	94	75	81	82	78
10000	97	92	76	81	81	78
12500	97	93	76	77	81	79
16000	93	92	75	73	80	77
20000	92	94	75	71	80	77
A-WEIGHTED LEVEL	118	115	104	99	109	107

REMARKS:

1. Runs one and two are without a muffler.
2. Runs three and four are with muffler design one.
3. Runs five and six are with the design one muffler modified by drilling four $\frac{1}{2}$" holes in the entrance pipe.

RUN NUMBER	7	8	9	10
SPACER SIZE(INCHES)	-	-	-	-
VALVE POSITION(TURNS)	5	5	$\frac{1}{4}$	$\frac{1}{4}$
INITIAL PRESSURE(PSIG)	2420	2080	1800	1580
FINAL PRESSURE(PSIG)	2080	1800	1580	1330
ELAPSED TIME(MIN)	5	5	7	5
AVE FLOW RATE(SCFM)	159	130	103	84
A-WEIGHTED LEVEL(dB)	113	116	112	106
LINEAR LEVEL-25 Hz (dB)	80	81	76	73
32	80	81	78	74
40	81	82	80	76
50	81	82	80	76
63	82	83	80	78
80	83	84	83	80
100	85	86	85	82
125	86	86	84	84
160	86	86	84	84
200	90	89	86	87
250	91	90	86	85
320	94	95	91	87
400	100	100	96	91
500	110	107	103	99
630	107	109	106	101
800	101	104	101	96
1000	98	98	95	93
1250	95	98	96	90
1600	94	98	96	88
2000	92	101	98	86
3200	86	99	95	77
4000	83	99	94	74
5000	82	95	91	73
6300	84	96	91	74
8000	83	97	93	73
10000	84	95	92	75
12500	82	94	93	75
16000	80	91	91	74
20000	78	89	91	73
A-WEIGHTED LEVEL	109	112	109	102

REMARKS:

1. Runs eight and nine are without a muffler.
2. Runs seven and ten are with muffler design two.

RUN NUMBER	11	12
SPACER SIZE(INCHES)	-	-
VALVE POSITION(TURNS)	$\frac{1}{4}$	$\frac{1}{4}$
INITIAL PRESSURE(PSIG)	1330	1180
FINAL PRESSURE(PSIG)	1180	1020
ELAPSED TIME (MIN)	5	5
AVE FLOW RATE (SCFM)	71	75
A-WEIGHTED LEVEL (dB)	104	109
LINEAR LEVEL-40 Hz (dB)	75	76
80	80	81
120	81	80
160	82	81
200	86	83
240	83	86
280	83	86
320	86	90
360	89	92
400	91	94
440	95	101
480	97	105
520	99	105
560	100	107
600	99	103
640	98	103
680	98	103
720	97	101
760	95	101
800	93	99
840	92	99
880	91	95
920	91	92
1000	90	91
A-WEIGHTED LEVEL	100	106

REMARKS:

1. Run eleven is with muffler design two.
2. Run twelve is without a muffler.

APPENDIX C

COMPUTER RESULTS

The muffler designs are analyzed using a computer program named MARTHA. MARTHA was developed at the Massachusetts Institute of Technology by Professor Paul Penfield, Jr. MARTHA uses APL (A Programmer's Language) to analyze linear electrical networks. One of the response functions generated by MARTHA is the transducer gain TG which is the ratio of P_{out} to P_{ga} and corresponds to the Insertion Loss of the network.

The computer input for each muffler design is entered into the computer first. Upon command, the computer prints out the Insertion Loss for the frequency spectrum selected. The values of Insertion Loss are then plotted against frequency.

```
      ⍝ THE FIRST COMMAND CALLS FOR THE COMPUTER
      ⍝ TO LOAD MARTHA

      )LOAD 100 MARTHA
SAVED  14.32.54 04/23/73

      ⍝ MUFFLER DESIGN ONE IS DEFINED

      ∇N←MUFFLER
[1]   N←(WS L1 S R1)WC(L2 S C1)WC(WS R2 S L3)WC(C2)
                WC(WS L4)WC(C3)WC(WS L5)

[2]   ∇

      ⍝ THE MUFFLER DESIGN IS TYPED ON ONE APL LINE, NOT
      ⍝ TWO AS SHOWN HERE

      ⍝ THE ELEMENT VALUES ARE ENTERED

      ZG←49400
      L1←L 36.4
      L2←L 17
      L3←L 127
      L4←L 135
      L5←L 63
      L6←L 2.85
]      R1←R 490
      R2←R 800
      R3←R 10000
      R4←R 20000
      C1←C 1.4E¯8
      C2←C 1.3E¯8
      C3←C 2E¯9
      C4←C 2E¯8

      ⍝ THE TERMINATION IMPEDANCE IS ENTERED

      ZL←L6 P ((R3 P C4) S R4)

      ⍝ THE FREQUENCY RANGE IS ENTERED

      F←40×⍳25
      TITLE←'   MUFFLER WITH ONE INCH.SPACERS+
                                            ^
                                            -DESIGN ONE'
      PRINT DB TG OF MUFFLER
```

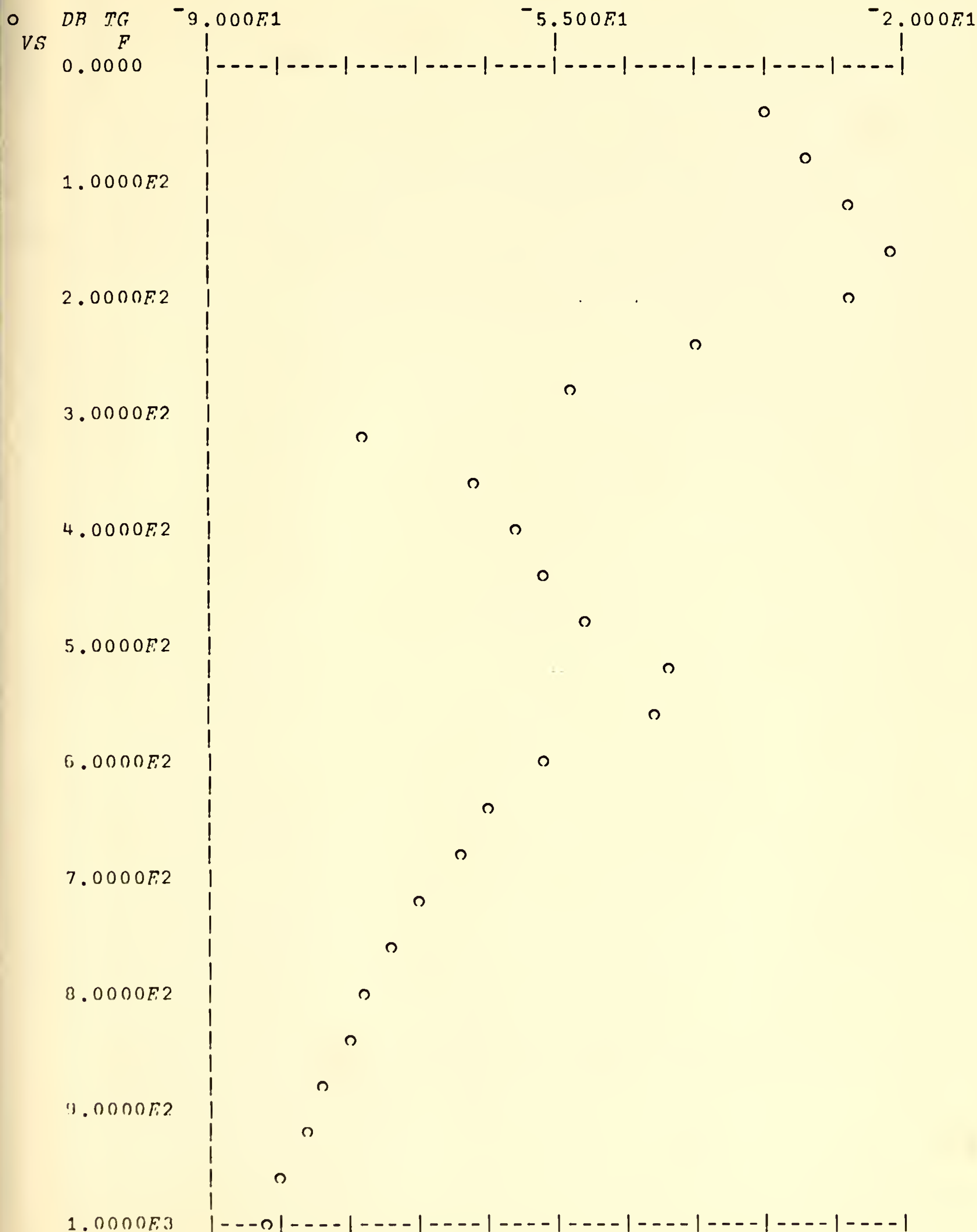
CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   10:5
MUFFLER WITH ONE INCH SPACERS-DESIGN ONE

F	DB TG
4.0000E1	¯3.3455E1
8.0000E1	¯2.9703E1
1.2000E2	¯2.4988E1
1.6000E2	¯2.1703E1
2.0000E2	¯2.6124E1
2.4000E2	¯4.0467E1
2.8000E2	¯5.3041E1
3.2000E2	¯7.5296E1
3.6000E2	¯6.3976E1
4.0000E2	¯5.9307E1
4.4000E2	¯5.6363E1
4.8000E2	¯5.2640E1
5.2000E2	¯4.4192E1
5.6000E2	¯4.5136E1
6.0000E2	¯5.5799E1
6.4000E2	¯6.1387E1
6.8000E2	¯6.5451E1
7.2000E2	¯6.8757E1
7.6000E2	¯7.1602E1
8.0000E2	¯7.4132E1
8.4000E2	¯7.6431E1
8.8000E2	¯7.8552E1
9.2000E2	¯8.0531E1
9.6000E2	¯8.2392E1
1.0000E3	¯8.4154E1

TITLE←'   MUFFLER WITH ONE INCH SPACERS-DESIGN ONE'
PLOT DB TG OF MUFFLER

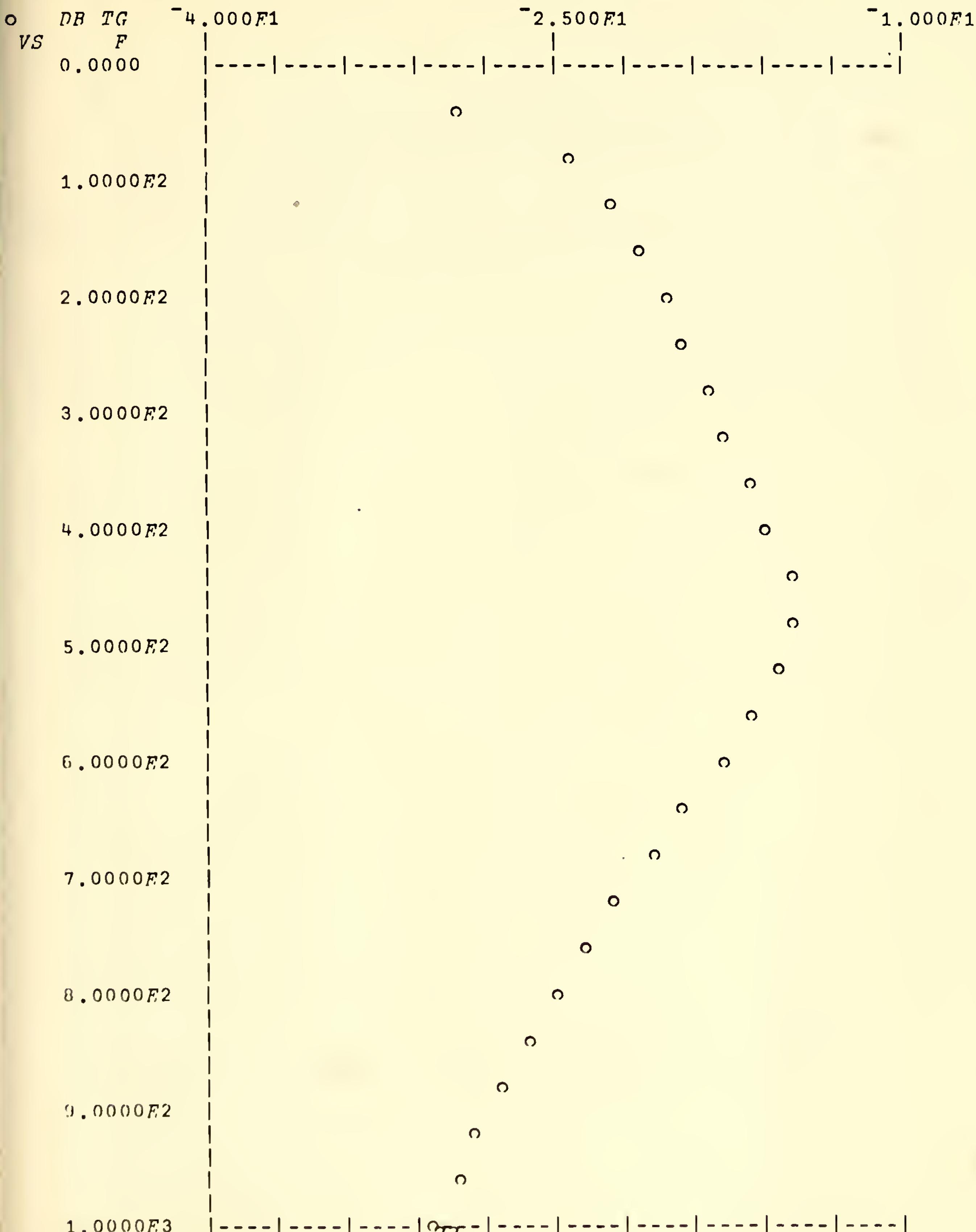
CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   10:8

MUFFLER WITH ONE INCH SPACERS-DESIGN ONE



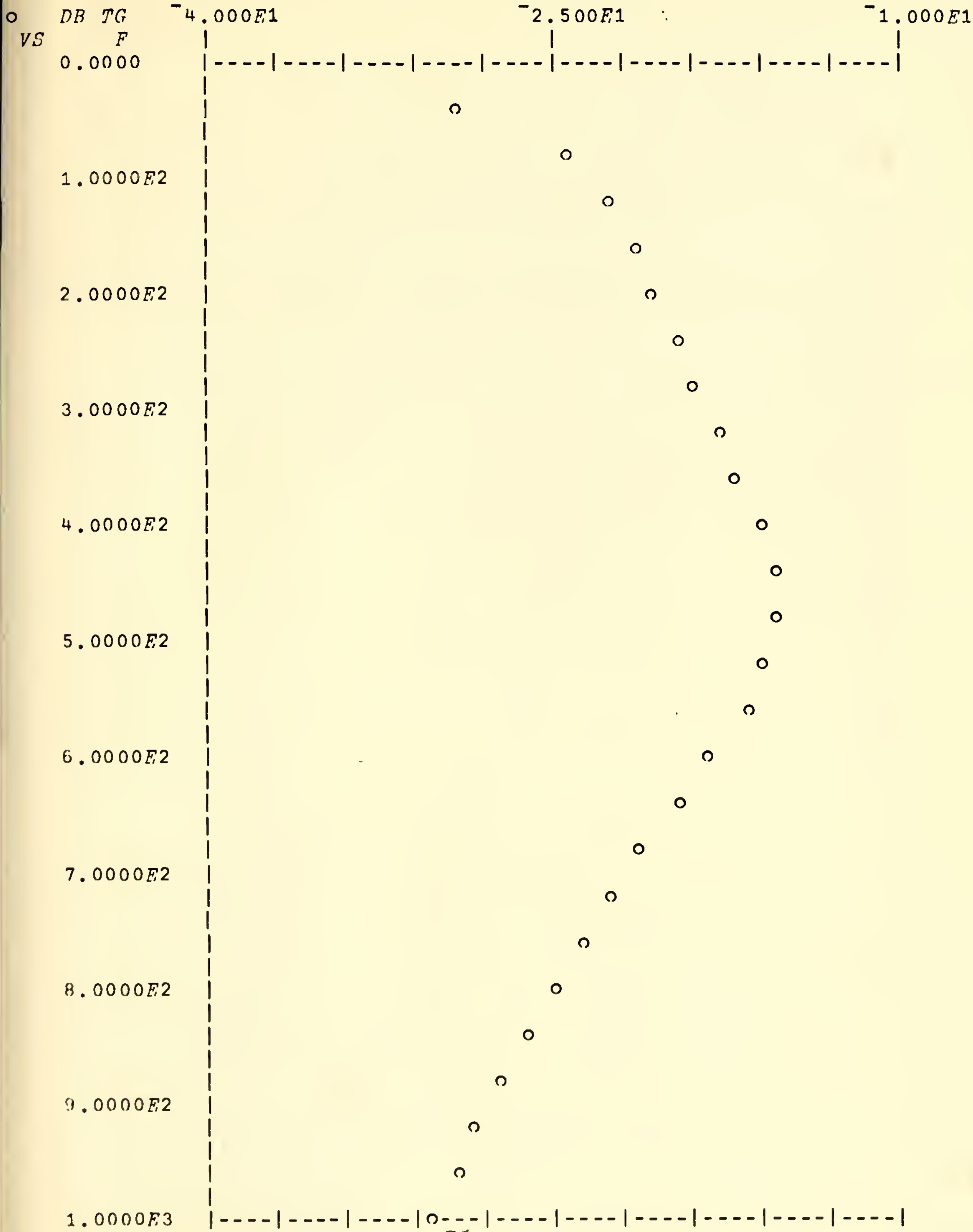
```
⍝ THE REVISED ANALOGY OF THE FIRST MUFFLER DESIGN
⍝ IS DEFINED

      ∇Q←MODONE
[1]    Q←(WS L1)WC(C1)WC(WS L2)
[2]    ∇
      L1←L 21
      C1←C 5.55E¯9
      L2←L 63

      TITLE←'    MUFFLER WITH ONE INCH SPACERS-REVISED ANALOGY'
      PRINT DB TG OF MODONE
CIRCUIT ANALYSIS BY MARTHA.    73∘A    4/27/73   10:18
   MUFFLER WITH ONE INCH SPACERS-REVISED ANALOGY

     F            DB TG
----------- -----------
 4.0000E1    ¯2.9183E1
 8.0000E1    ¯2.4519E1
 1.2000E2    ¯2.2475E1
 1.6000E2    ¯2.1263E1
 2.0000E2    ¯2.0336E1

 2.4000E2    ¯1.9483E1
 2.8000E2    ¯1.8614E1
 3.2000E2    ¯1.7694E1
 3.6000E2    ¯1.6734E1
 4.0000E2    ¯1.5809E1

 4.4000E2    ¯1.5098E1
 4.8000E2    ¯1.4869E1
 5.2000E2    ¯1.5326E1
 5.6000E2    ¯1.6400E1
 6.0000E2    ¯1.7825E1

 6.4000E2    ¯1.9363E1
 6.8000E2    ¯2.0884E1
 7.2000E2    ¯2.2332E1
 7.6000E2    ¯2.3693E1
 8.0000E2    ¯2.4969E1

 8.4000E2    ¯2.6168E1
 8.8000E2    ¯2.7297E1
 9.2000E2    ¯2.8367E1
 9.6000E2    ¯2.9385E1
 1.0000E3    ¯3.0357E1

      TITLE←'    MUFFLER WITH ONE INCH SPACERS-REVISED ANALOGY'
      PLOT DB TG OF MODONE
```

CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   10:20
MUFFLER WITH ONE INCH SPACERS-REVISED ANALOGY



```
⍝ THE MUFFLER IS TUNED BY CHANGING THE SPACERS

      R3←R 20000
      R4←R 40000
      L6←L 4.0
      C4←C 7E¯9
      TITLE←'    MUFFLER WITH HALF INCH SPACERS-REVISED ANALOGY'
      ZL←L6 P ((R3 P C4) S R4)
      PRINT DB TG OF MODONE
CIRCUIT ANALYSIS BY MARTHA.    73∘A    4/27/73   10:32
   MUFFLER WITH HALF INCH SPACERS-REVISED ANALOGY

      F            DB TG
----------- -----------
 4.0000E1    ¯2.9267E1
 8.0000E1    ¯2.4638E1
 1.2000E2    ¯2.2625E1
 1.6000E2    ¯2.1439E1
 2.0000E2    ¯2.0534E1

 2.4000E2    ¯1.9702E1
 2.8000E2    ¯1.8852E1
 3.2000E2    ¯1.7952E1
 3.6000E2    ¯1.7011E1
 4.0000E2    ¯1.6109E1

 4.4000E2    ¯1.5424E1
 4.8000E2    ¯1.5224E1
 5.2000E2    ¯1.5702E1
 5.6000E2    ¯1.6779E1
 6.0000E2    ¯1.8192E1

 6.4000E2    ¯1.9706E1
 6.8000E2    ¯2.1194E1
 7.2000E2    ¯2.2603E1
 7.6000E2    ¯2.3918E1
 8.0000E2    ¯2.5142E1

 8.4000E2    ¯2.6282E1
 8.8000E2    ¯2.7347E1
 9.2000E2    ¯2.8348E1
 9.6000E2    ¯2.9291E1
 1.0000E3    ¯3.0185E1

      TITLE←'    MUFFLER WITH HALF INCH.SPACERS-REVISED ANALOGY'
      PLOT DB TG OF MODONE
```

CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   10:34
MUFFLER WITH HALF INCH.SPACERS-REVISED ANALOGY



```
R3←R 6600
R4←4
    V
   R 13000
L6←L 2.3
C4←C 3.7E¯8
ZL←L6 P ((R3 P C4) S R4)
TITLE←'   MUFFLER WITH ONE AND ONE HALF INCH SPACERS-
         REVISED ANALOGY'
PRINT DB TG OF MODONE
```

```
CIRCUIT ANALYSIS BY MARTHA.    73∘A    4/27/73   10:40
   MUFFLER WITH ONE AND ONE HALF INCH SPACERS-
              REVISED ANALOGY
```

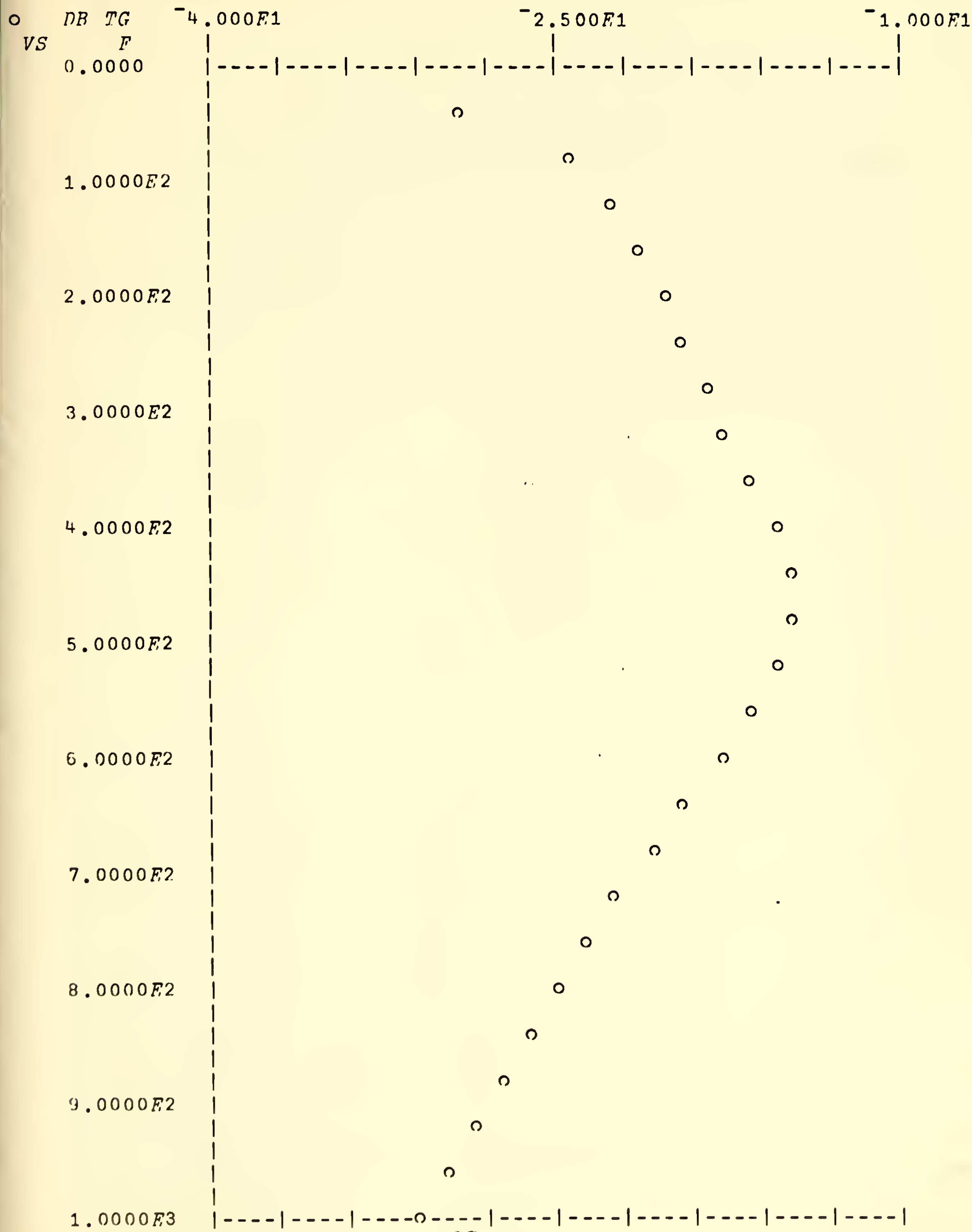
F	DB TG
4.0000E1	¯2.9187E1
8.0000E1	¯2.4504E1
1.2000E2	¯2.2441E1
1.6000E2	¯2.1212E1
2.0000E2	¯2.0269E1
2.4000E2	¯1.9402E1
2.8000E2	¯1.8521E1
3.2000E2	¯1.7592E1
3.6000E2	¯1.6624E1
4.0000E2	¯1.5695E1
4.4000E2	¯1.4980E1
4.8000E2	¯1.4752E1
5.2000E2	¯1.5218E1
5.6000E2	¯1.6313E1
6.0000E2	¯1.7773E1
6.4000E2	¯1.9357E1
6.8000E2	¯2.0929E1
7.2000E2	¯2.2435E1
7.6000E2	¯2.3859E1
8.0000E2	¯2.5202E1
8.4000E2	¯2.6470E1
8.8000E2	¯2.7671E1
9.2000E2	¯2.8813E1
9.6000E2	¯2.9903E1
1.0000E3	¯3.0946E1

```
TITLE←'   MUFFLER WITH ONE AND ONE HALF INCH SPACERS-
             REVISED ANALOGY'
PLOT DG
      V
      B TG OF MODONE
```

CIRCUIT ANALYSIS BY MARTHA. 73∘A 4/27/73 10:43

MUFFLER WITH ONE AND ONE HALF INCH SPACERS-

REVISED ANALOGY

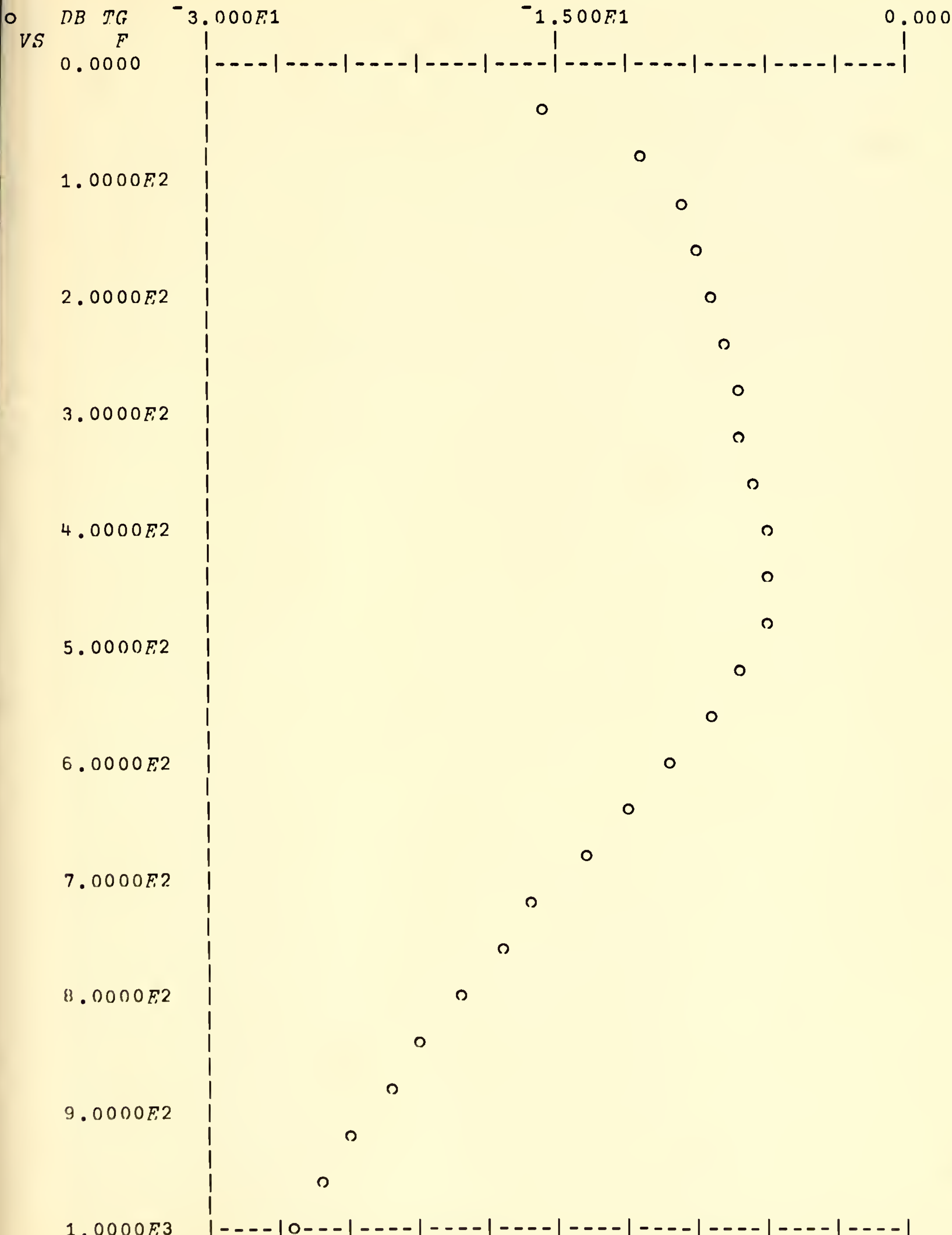


```
    ⍝ THE EFFECT OF AIR FLOW IS CONSIDERED FOR MUFFLER
    ⍝ DESIGN ONE WITH ONE INCH SPACERS-REVISED ANALOGY
    L6←L 14.3
    R3←R 10000
    R4←R 20000
    C4←C 2E¯8
    ZL←L6.
          v
          P ((R3 P C4) S R4)
    TITLE←'    MUFFLER DESIGN ONE-REVISED ANALOGY
  TERMINATION IMPEDANCE VALUE HANGE(L6=14.3)
'
    PRINT DB TG OF MODONE

CIRCUIT ANALYSIS BY MARTHA.     73∘A    4/27/73   10:54
   MUFFLER DESIGN ONE-REVISED ANALOGY
   TERMINATION IMPEDANCE VALUE HANGE(L6=14.3)
```

F	DB TG
4.0000E1	¯1.5448E1
8.0000E1	¯1.1292E1
1.2000E2	¯9.7051
1.6000E2	¯8.8953
2.0000E2	¯8.3586
2.4000E2	¯7.9133
2.8000E2	¯7.4834
3.2000E2	¯7.0396
3.6000E2	¯6.5911
4.0000E2	¯6.2011
4.4000E2	¯6.0144
4.8000E2	¯6.2507
5.2000E2	¯7.0871
5.6000E2	¯8.4953
6.0000E2	¯1.0265E1
6.4000E2	¯1.2174E1
6.8000E2	¯1.4084E1
7.2000E2	¯1.5927E1
7.6000E2	¯1.7677E1
8.0000E2	¯1.9331E1
8.4000E2	¯2.0892E1
8.8000E2	¯2.2366E1
9.2000E2	¯2.3762E1
9.6000E2	¯2.5085E1
1.0000E3	¯2.6343E1

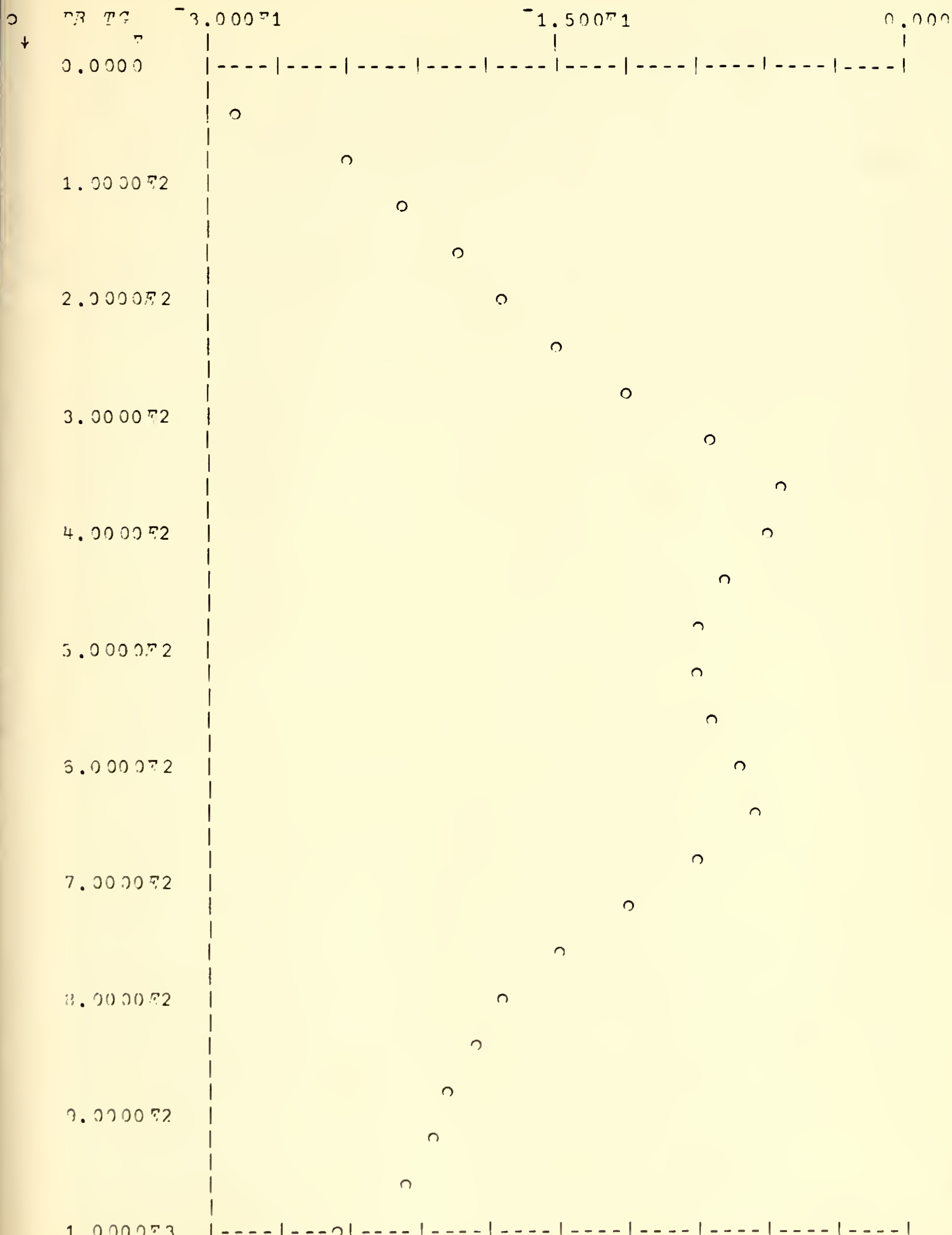
```
    TITLE←'    MUFFLER DESIGN ONE-REVISED ANALOGY
  TERMINATION IMPEDANCE VALUE CHANGE(L6=14.3)
'   PLOT DB TG OF MODONE
```

CIRCUIT ANALYSIS BY MARTHA.   73.A   4/27/73   10:58
MUFFLER DESIGN ONE-REVISED ANALOGY
TERMINATION IMPEDANCE VALUE CHANGE(L6=14.3)



```
⍝ THE SECOND MUFFLER DESIGN
⍝ THE SPACERS ARE ALWAYS ONE INCH LONG


     ∇P←TWO
[1]  P←(WS L8)WC(C5)WC(WS L9)WC(C6)WC(WS L10)WC(C7)WC(WS L11)
[2]  ∇
     L8←L 16.3
     L9←L 25.2
     L10←L 25.2
     L11←L 16.0
     C5←C 3E¯9
     C6←C 4.4E¯9
     C7←C 9E¯9
     R3←R 10000
     R4←R 20000
     L6←L 2.85
     C4←C 2E¯8
     ZL←L6 P ((R3 P C4) S R4)
     TITLE←'   MUFFLER DESIGN TWO'
     PRINT DB TG OF TWO


CIRCUIT ANALYSIS BY MARTHA.    73∘A    4/27/73   11:6
   MUFFLER DESIGN TWO

     F            DB TG
----------- -----------
 4.0000E1    ¯2.9049E1
 8.0000E1    ¯2.4020E1
 1.2000E2    ¯2.1400E1
 1.6000E2    ¯1.9370E1
 2.0000E2    ¯1.7319E1

 2.4000E2    ¯1.4933E1
 2.8000E2    ¯1.1985E1
 3.2000E2    ¯8.4175
 3.6000E2    ¯5.4147
 4.0000E2    ¯5.7730

 4.4000E2    ¯7.6983
 4.8000E2    ¯8.9806
 5.2000E2    ¯9.2986
 5.6000E2    ¯8.6641
 6.0000E2    ¯7.2709

 6.4000E2    ¯6.5163
 6.8000E2    ¯8.7107
 7.2000E2    ¯1.2164E1
 7.6000E2    ¯1.5088E1
 8.0000E2    ¯1.7238E1

 8.4000E2    ¯1.8688E1
 8.8000E2    ¯1.9575E1
 9.2000E2    ¯2.0272E1
 9.6000E2    ¯2.1756E1
 1.0000E3    ¯2.4878E1
```

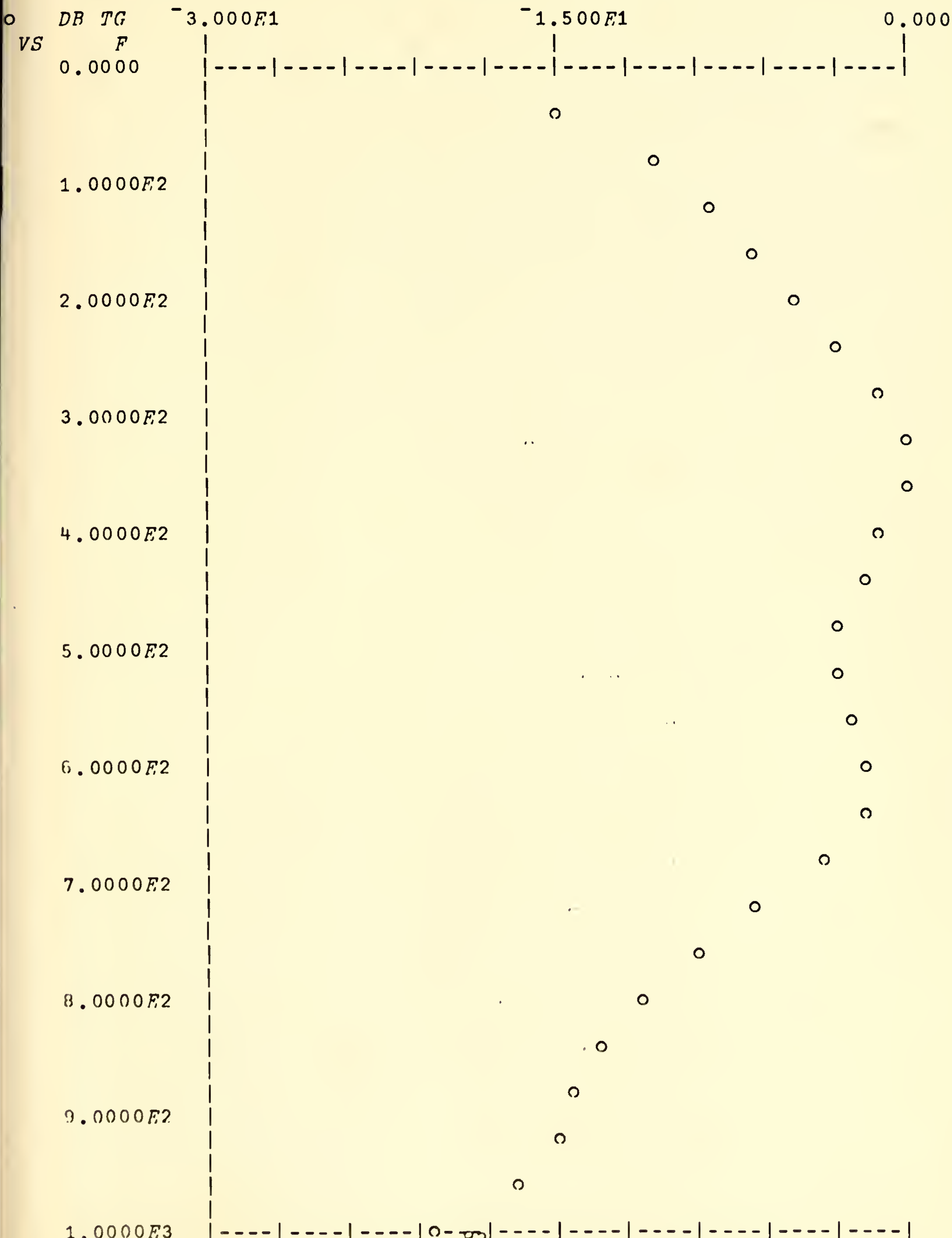
CIRCUIT ANALYSIS BY MARTHA. 730X 4/9/73 11:41
MUFFLER DESIGN TWO



```
⍝ THE EFFECT OF AIR FLOW IS CONSIDERED FOR MUFFLER
⍝ DESIGN TWO

L6←L 14.3
R3←R 10000
R4←R 20000
C4←C 2E¯8
ZL←L6 P ((R3 P C4) S R4)
TITLE←'   MUFFLER DESIGN TWO
  TERMINATION IMDEPANCE VALUE CHANGE(L6=14.3)
'
PRINT DB TG OF TWO

CIRCUIT ANALYSIS BY MARTHA.    73∘A    4/27/73   11:14
   MUFFLER DESIGN TWO
   TERMINATION IMDEPANCE VALUE CHANGE(L6=14.3)
```

F	DB TG
4.0000E1	¯1.5254E1
8.0000E1	¯1.0612E1
1.2000E2	¯8.3212
1.6000E2	¯6.5736
2.0000E2	¯4.8535
2.4000E2	¯3.0324
2.8000E2	¯1.2792
3.2000E2	¯1.5644E¯1
3.6000E2	¯2.0429E¯1
4.0000E2	¯1.0877
4.4000E2	¯2.0664
4.8000E2	¯2.7088
5.2000E2	¯2.8444
5.6000E2	¯2.4323
6.0000E2	¯1.6821
6.4000E2	¯1.5849
6.8000E2	¯3.4399
7.2000E2	¯6.4436
7.6000E2	¯9.2855
8.0000E2	¯1.1554E1
8.4000E2	¯1.3210E1
8.8000E2	¯1.4338E1
9.2000E2	¯1.5288E1
9.6000E2	¯1.7019E1
1.0000E3	¯2.0382E1

CIRCUIT ANALYSIS BY MARTHA.   73∘A   4/27/73   11:17
MUFFLER DESIGN TWO
TERMINATION IMPEDANCE VALUE CHANGE (L6=14.3)



Thesis 145776
M886 Mulholland
Suppression of hyper-
baric chamber noise.